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A review of heat pump water heating systems

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ABSTRACT

A heat pump water heater (HPWH) operates on an electrically driven vapor-compression cycle and pumps energy from the air in its surroundings to water in a storage tank, thus raising the temperature of the water. HPWHs are a promising technology in both residential and commercial applications due to both improved efficiency and air conditioning benefits.

Residential HPWH units have been available for more than 20 years, but have experienced limited success in the marketplace. Commercial-scale HPWHs are also a very promising technology, while their present market share is extremely low.

This study dealt with reviewing HPWH systems in terms of energetic and exergetic aspects. In this context, HPWH technology along with its historical development was briefly given first. Next, a comprehensive review of studies conducted on them were classified and presented in tables. HPWHs were then modeled for performance evaluation purposes by using energy and exergy analysis methods. Finally, the results obtained were discussed. It is expected that this comprehensive review will be very beneficial to everyone involved or interested in the energetic and exergetic design, simulation, analysis, performance assessment and applications of various types of HPWH systems.

Contents

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Abbreviations: A/C, air/conditioning; ASHP, air-source heat pump; ASHPWH, air-source heat pump water heater; CACS, conventional air-conditioning system; CHP, combined heat and power; DHW, domestic hot water; DX-SAHP, direct-expansion solar-assisted heat pump; DX-SAHPWH, direct-expansion solar-assisted heat pump water heater; EF, energy factor; EHP, electric driven heat pump; EUS, energy utilization systems; GEHP, gas engine driven heat pump; GSHP, ground-source heat pump; HP, heat pump; HPSAHP, heat-pipe enhanced solar-assisted heat pump water heater; HPWH, heat pump water heater; HVAC, heating ventilating and air conditioning; ISAHP, integral type solar-assisted heat pump water heater; PV, photovoltaic; SAS-HPWH, solar-air source heat pump water heater; SF, solar fraction; WH, water heater; WLHPS, water-loop heat pump system; ZNEH, zero net energy homes.

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Nomenclature

A	surface area (m^2)
C	specific heat (kJ/kg K)
COP	heating coefficient of performance
$\cos\varphi$	power factor
$\dot{E}$	energy rate (kW)
$\dot{E}x$	exergy rate (kW)
f	friction factor or exergetic factor (%)
$\dot{F}$	exergetic fuel rate (kW)
h	specific enthalpy (kJ/kg)
I	current (A), global irradiance (W/m^2)
$I\dot{P}$	improvement potential rate (kW)
$\dot{I}$	irreversibility (exergy destruction) rate (kW)
$\dot{m}$	mass flow rate (kg/s)
P	pressure (kPa)
$\dot{P}$	exergetic product rate (kW)
s	specific entropy (kJ/kg K)
$\dot{S}$	entropy rate (kW/K)
$\dot{Q}$	heat transfer rate (kW)
T	temperature (K or °C)
V	voltage (V)
$\dot{W}$	rate of work or power (kW)

Greek letters

δ	fuel depletion rate (%)
ε	exergy (second law) efficiency
η	energy efficiency
ξ	productivity lack (%)
χ	relative irreversibility (%)
ψ	specific exergy (kJ/kg)

Subscripts

act	actual
avg	averaged
c	space cooling
coll	collector
comp	compressor
cond	condenser
cw	space cooling and water heating
dest	destroyed (destruction)
dhwt	domestic hot water tank
e	evaporation
elec	electrical
evap	evaporator
exp	expansion (throttling) valve
fc	fan-coil
fh	floor heating
fhs	floor heating system
gen	generation
ghe	ground heat exchanger
GSHP	ground-source heat pump
h	heating
HP	heat pump
i	each unit value
in	input, inlet
k	location
mech	mechanical
out	outlet, output
p	pressure
r	refrigerant
scol	solar collector
SDHWS	solar domestic hot water system
sh	space heating
sr	solar radiation
t	thermal
T	total
u	useful
w	water
over dot	rate
0	reference (dead) state

1. Introduction

Water heating is the fourth largest energy user in the commercial buildings sector, after heating, air conditioning, and lighting. It is a major energy user in building types such as full-service restaurants, motels and hotels, assisted living centers, and other facilities that do a great deal of laundry or dishwashing [1].

Water heating accounts for 17% of all residential site energy use in the United States, making it the third largest use of energy in homes. That percentage varies somewhat, with the average home in some states (e.g., California) using 25% of its energy for water heating. Nearly 40% of all homes in the U.S. use electricity to heat water. Again, that fraction varies considerably from state to state, with more than 80% of homes in some states (e.g., Florida) and less than 15% of others (e.g., California and New York) using electric water heaters (WHs) [2].

Most residential WHs are equipped with conventional heaters generating heat by consuming fossil fuels or electricity. Those WHs are usually simple, but not desirable in view of energy utilization efficiency. For instance, electric WHs are convenient for installation and operation, however, the overall efficiency in converting a potential energy of fossil fuels into electric energy, then into thermal energy is quite low. Compared to those WHs, heat pump (HP) water heating systems can supply much more heat just with the same amount of electric input used for conventional heaters [3].

HP systems are heat-generating devices that can be used to heat water to be used in either domestic hot water or space heating applications. For HP, a basic factor of great importance for its successful application is the availability of a cheap, dependable heat source for the evaporator-preferably one at relatively high temperature. The coefficients of performance (COP) of HP systems depend on many factors, such as the temperature of low-energy source, the temperature of delivered useful heat, the working medium used, the characteristics of components of HP systems, etc. Among the above mentioned, the temperature of the evaporator is a key factor [4].

Since the 1950s, researches have been performed on heat pump water heaters (HPWHs), including structure, thermodynamics, working fluids, operation controlling, numerical simulation and economical analysis [5–7].

Kim et al. [3] designed a dynamic model of a WH system driven by a HP to investigate transient thermal behavior of the system which was composed of a HP and a hot water circulation loop. From the simulation, the smaller size of the water reservoir was found to have larger transient performance degradation, and the larger size caused additional heat loss during the hot water storage period. Therefore, the reservoir size should be optimized in a

design process to minimize both the heat loss and the performance degradation.

Laipradit et al. [8] investigated theoretical performance analysis of HPWHs using carbon dioxide as refrigerant. For rated capacities of a 4 kW compressor with a 10 kW gas cooler and a 6 kW evaporator, the coefficient of performance is found to be between 2.0 and 3.0. The mass flow rate ratio of water and CO_2 between 1.2 and 2.2 is the most suitable value for generating hot water temperature above 60 °C at 15–25 °C ambient air temperature.

Rankin et al. [9] presented a study about demand side management for commercial building using an inline HPWH methodology. Rousseau and Greyvenstein [10] also performed enhancing the impact of HPWH in the South African commercial sector.

The main objectives in doing the present study are: (i) to review studies conducted on HPWH systems taking into consideration the analysis that have been made (theoretical–experimental, energy–exergy) and the utilization of the systems (space heating, water heating, air conditioning (A/C)), (ii) to present a mathematical model for exergy-based HPWH calculations including thermodynamic parameters, such as fuel depletion ratio, relative irreversibility, productivity lack and exergetic factor as well as improvement potential, and (iii) to apply the present mathematical model to an illustrative example.

2. A brief description of HPWH technology

HPWHs are a promising technology and use the same mechanical principles as refrigerators and air conditioners. While refrigerators remove heat from the interior and discharge it to the (kitchen) environment, HPWHs take heat from the environment and concentrate it to heat water for service needs [1].

A HP is a machine that transfers heat from a source to other by employing a refrigeration cycle. Although heat normally flows from higher to lower temperatures, a HP reverses that flow and acts as a "pump" to move the heat. Therefore, a HP can be used both for space heating in the winter and for cooling (air conditioning) in the summer. In the refrigeration cycle, a refrigerant (known as the "working fluid") is compressed (as a liquid) then expanded (as a vapor) to absorb and remove heat. The HP transfers heat to a space to be heated during the winter period and by reversing the operation, extracts (absorbs) heat from the same space to be cooled during the summer period [11].

Fig. 1 illustrates the schematic of a simplified HPWH system, while the current status of the HPWH technology, its savings potential, its potential financial attractiveness to consumers, and market barriers to significant penetration into the market have been given in more detail in Ref. [2].

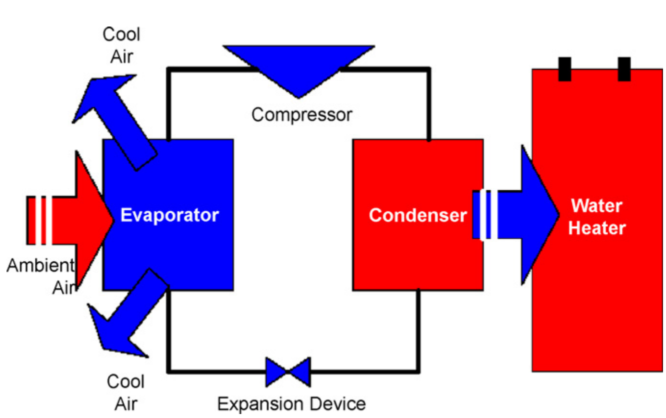

Fig. 1. Schematic of a simplified HPHW system [2].

The HPWH removes heat from the ambient air and puts it into a water storage tank. This process produces cooler, dryer (depending on relative humidity) exhaust air than the source air. Therefore, placement of the HPWH relative to the conditioned space affects overall building energy use. A HPWH has an easier time generating 60 °C water if the ambient air is 24 °C than if it is 1.7 °C. Once the ambient temperature drops below a certain threshold (depending on the configuration of the unit and the refrigerant, this is typically around 0 °C), the HP can no longer extract meaningful heat, and an electric resistance back-up element is employed. As the temperature approaches this threshold, the condenser must go through defrost cycles to clear accumulated ice, and this lowers efficiency. Additionally, HPWHs typically have a longer recovery rate than electric resistance units, which necessitates larger tank size.

HPWHs can be used for heat water only or multi-functional purposes. Many researchers have presented different systems and analyzed their efficiencies. For example, these systems include air-source heat pumps (ASHPs), ground-source heat pumps (GSHP), solar-assisted heat pumps, direct-expansion solar-assisted heat pump (DX-SAHPs), integrated solar-assisted HPs, gas engine driven heat pumps (GEHPs) and multi-function HPs.

As for as efficiencies of HPWHs are concerned, HPWHs have an energy factor (EF) of between 2 and 3, depending on the unit (EF is used to rate the energy efficiency of storage-type hot WHs; the higher the EF, the more efficient the unit). This suggests savings relative to standard new electric resistance WHs (whose EFs are typically close to 0.9) of 55–70%. However, multiple field test data indicate that actual water heating energy savings relative to standard electric resistance WHs are 40–60%. There are several factors that contribute to this, including recharge rate, defrost cycles, and inlet water and ambient air temperatures. As a result of the fact that the HPWH uses ambient air to heat the water and then exhausts cooler and dryer air, the placement of the unit within the conditioned space has an impact on the building's heating, cooling, and dehumidification energy use [2].

3. A brief historical development of heat pumps and HPWHs

The use of HPs for the heating and cooling of the Equitable Building, initiated in 1948, was a pioneering achievement in the Western hemisphere. The theoretical conception of the HP was described in a neglected book, published in 1824, and written by a young French army officer, Sadi Carnot. Its practical application on a large scale is attributable to designers J. Donald Kroeker and Ray C. Chewning; building engineer, Charles E. Graham, and architect Pietro Belluschi [12].

The tremendously gifted Irish scientist William Thomson (Lord Kelvin) is usually credited with the concept of the HP, though he did not have the resources to construct one. The first patent for this device was awarded in 1927 to T.G.N. Haldane, an English inventor [13].

In 1948 if one were to predict the site of the first commercial HP installation in the US, Portland, Oregon would probably not even be on the list. Located in the north end of the Willamette Valley, the confluence of the Willamette River and the Columbia River, Portland enjoys a moderate climate. Winter average temperature is approximately 38°, while the summer average is 64 °F. In 1948 the standard for building design because of the moderate temperatures in the summertime, did not require air conditioning in the Willamette Valley [12].

Commercial distribution of HP units, which was initiated in the early 1950s, suffered declines in the 1960s due to a record of poor reliability, but registered rapid growth after 1970, when higher electricity costs made electric furnaces less competitive, and improved quality control increased the attractiveness of HPs.

The oil crisis in the beginning of the 1970s has led researchers to use alternative energy sources for energy production [4]. Later, HPs became popular for heating and cooling applications and ground source, air source, combining solar energy and GEHPs were proposed by many researchers [14].

In 1996, the world's largest installation of geothermal HPs was completed at the U.S. Army's Fort Polk military base in Leesville, Louisiana. The HPs replaced 3243 ASHPs and 760 central air conditioning/natural gas forced air furnace systems for 4003 housing units. The housing units were apartments, townhouses, and duplexes built between 1972 and 1988. Unit floor space ranged from about 84 to 130 m^2. The geothermal HP configuration implemented at Fort Polk is a closed-loop, vertical-borehole ground heat exchanger system. Each HP has its own ground heat exchanger of the vertical U-tube type of polyethylene pipe. Over 8000 borehole heat exchangers were drilled. Each borehole has a 4-in. diameter and a depth of about 30–137 m [15].

HPs, involving higher initial costs than electric furnaces, are attractive as long as their efficiency (200% or more) provides an eventual return on the investment. Solar heating systems, with high seasonal efficiencies (400–600%), nevertheless may not be attractive investments if initial costs are very high [14].

4. Reviewing and classifying studies conducted on HPWHs

In the following section, various types of HPWHs are reviewed and classified in terms of applications, including some experimental data on operating and performance.

4.1. Reviewing studies conducted

Swardt and Meyer [16] studied on the performance of a reversible GSHP coupled to a municipality water reticulation system, compared experimentally and with simulations to a conventional ASHP for space cooling and heating. GSHPs have demonstrated the potential to reduce peak demands and total electricity consumption. The use of a municipality water reticulation system as a heat source/sink resulted in an annual improvement of 13% in capacities and an annual improvement of 14% in COPs. Especially at low ambient air temperatures, GSHPs have significant heating capacity (24%) and efficiency improvements (20%) over ASHPs. The schematic diagram is shown in Fig. 2.

Hepbasli et al. [17] performed for the first time in Turkey at the university level the performance characteristics of a GSHP system with a 50 m vertical 11/4 in. nominal diameter U-bend ground heat exchanger. This system was installed in a 65-m^2 room in the Solar Energy Institute, Ege University, Izmir (568 degree days cooling, base: 22 °C, 1226 degree days heating, base: 18 °C), Turkey. The heating and cooling loads of the room were 3.8 and 4.2 kW at design conditions, respectively. The system was commissioned in May 2000 and performance tests have been conducted since then. Based upon the measurements made in the heating mode, the heat extraction rate from the soil, with an average thermal diffusivity of 0.00375 m^2/h, was found to be, on average, 11 W/m of bore depth, while the required borehole length in meter per kW of heating capacity was obtained to be 14.7. The entering water temperature to the unit ranged from 5.5 to 13.2 °C, with an average value of 8 °C. The heating coefficient of performance of the HP and the whole system was extremely low when compared to other HPs operating under conditions at or near design values (Fig. 3).

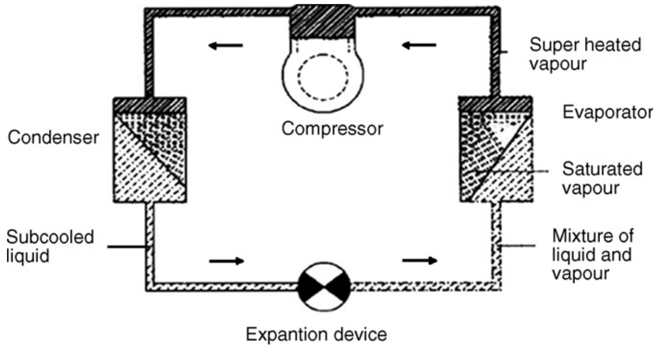

Fig. 2. Cycle layout [16].

Various means of producing domestic hot water (DHW) with renewable energy in zero net energy homes (ZNEH) were examined for two climates (Montréal and Los Angeles) by Biaou and Bernier [18]. Four alternatives investigated included (i) a regular electric hot water tank; (ii) the desuperheater of a GSHP with electric backup; (iii) thermal solar collectors with electric backup; and (iv) a HPWH indirectly coupled to a space conditioning GSHP, as can be seen in Fig. 4. Results indicated that heating DHW with thermal solar collectors with an electric backup (which is either provided by the photovoltaic (PV) panels or the grid in a ZNEH) was the best solution for a ZNEH. The second part of this paper focused on determining what should be the respective areas of the thermal solar collectors and PV array to obtain the least expensive solution to achieve total DHW production with renewable energy.

The most common type of HP for domestic use, referred to as a "conventional" HP, is the air-to-air (air source) system in which heat is taken from air (heat source) at one location and transferred to air (heat sink) at another location. In the winter, a HP takes heat from outside air and via a working fluid transports the heat to inside the home. When the outside air temperature drops below −3.88, −1.1 °C, the ASHP uses electric resistance heat. In the summer, the HP reverses the process, removing heat from the home and transporting it to outside air, cooling the home in the process [11]. There have been many studies conducted on ASHP.

One of them, a pilot run facility for assembly and testing of 100 units was designed and constructed. After assembly and shipment, these 100 units were placed in customers' homes by 20 electric utility companies in order to demonstrate the capabilities of the HPWH. The performance of the units was being monitored for 1 year by these utilities; the data would be forwarded to energy utilization systems (EUS) and would be presented in a final report by Harris et al. [2] and Sloane et al. [19]. Two HPWH models have been developed. One model was designed as a complete unit, consisting of a HP assembly mounted at top a 0.31-m^3 water tank. The second model was a retrofit unit, designed to be used with an existing WH installation. The first model is demonstrated in Fig. 5. The COP values ranged from about 2.0 at 18 °C ambient air and 27 °C source water, to about 2.8 at 35 °C ambient air and 4.4 °C source water.

Harata et al. [20] used thermoelectric technology. The thermoelectric technology changing electric energy into the heat energy was commonly known to be used as the HP. In this thermoelectric technology, they used a HP utilizing the latent heat of the atmosphere. The other characteristic was to collect the heat occurring from the power supply equipment of the thermoelectric technology device in effect a combined heat and power (CHP) device. In this thermoelectric technology, an improvement of the energy efficiency could be expected. Initially they examined for a basis a bench scale model. As for examination condition, the capacity of the tank was 3.7 l, the temperature of hot water was 85 °C, the environment temperature was 23 °C and the efficiency of the power supply was 80% (Fig. 6).

Ito and Miura [21] investigated the mechanism of HPs for hot water supply using dual heat sources of the ambient air and water

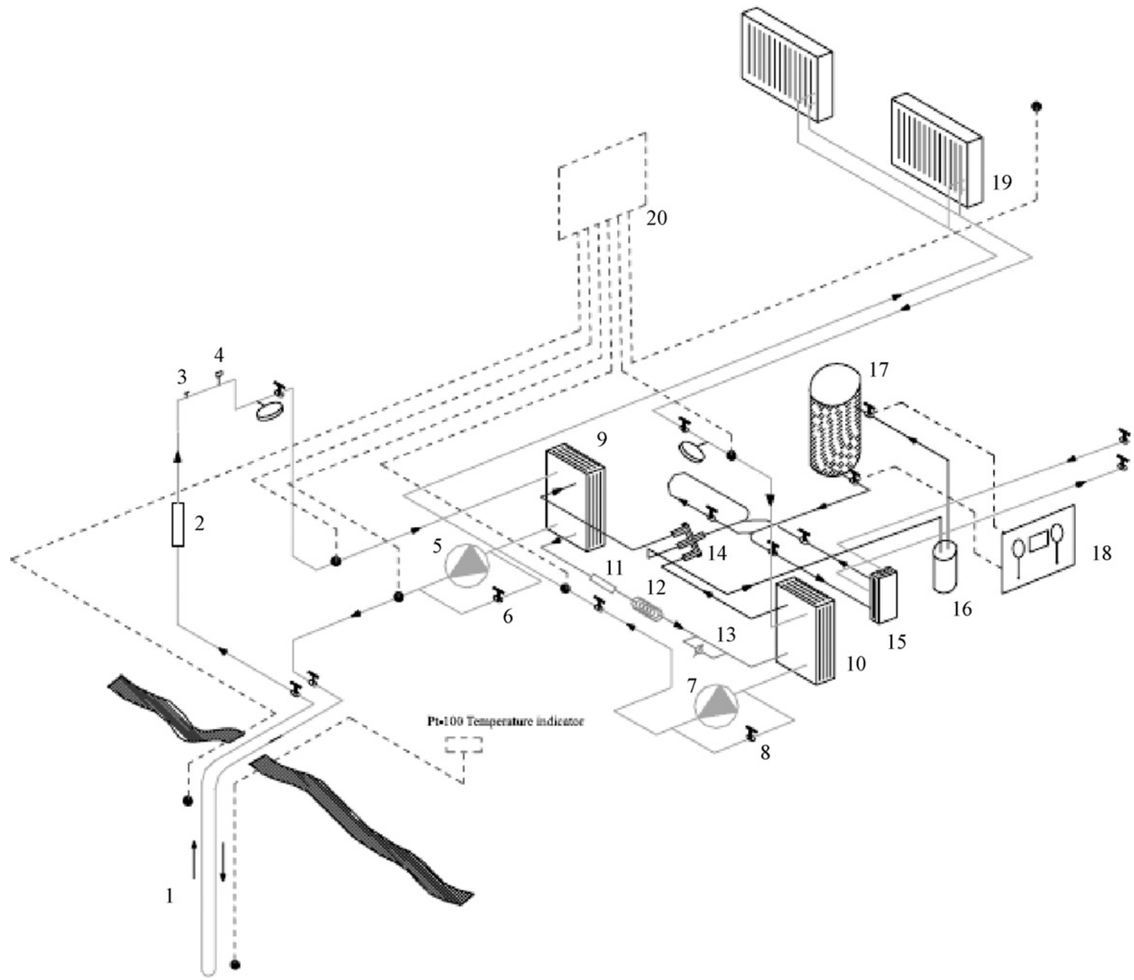

Fig. 3. Isometric stand: (1) ground heat exchanger; (5) brine circulating pump; (7) water circulating pump; (9) heat exchanger; (10) heat exchanger; (12) capillary tube; (15) desuperheater; (17) compressor; (19) fan-coil units [17].

and the operating conditions of selecting either one or both heat sources. Then, the performance of a HP, which used water and the ambient air as the heat sources to heat water, was experimentally studied. When the temperature of the water heat source was decreased, the heat from the water as well as the heat from the air was used for the HP efficiently until its temperature became approximately that of the evaporation temperature of the HP using the ambient air alone as the heat source. When the temperature of the water dropped further, only the heat from the air was absorbed by the evaporators like an ordinary HP, which used only the evaporator of the air heat source at the same ambient air temperature. The system is illustrated in Fig. 7.

Ji et al. [22] introduced a novel air-conditioning product, that could achieve the multi-functions with improved energy performance. They reported the basic design principles and the laboratory test results. The results showed that by incorporating a WH in the outdoor unit of a split-type air-conditioner, so that space cooling and water heating could take place simultaneously, the energy performance could be raised considerably. Two prototypes of slightly different design were fabricated for performance testing (Figs. 8 and 9). Averaged COP, for space cooling and water heating, water heating only and space heating only, was obtained 4.02, 2.91, 2.00 (ambient temperature at 4.5 °C) and 2.72, respectively.

One the other study, seasonal performance evaluation methods for WHs was reviewed, while an experimental method for rating ASHPWHs was presented by Morrison et al. [23]. The rating method was based on the measured HP performance during the heat-up operation of particular products rather than a generic simulation model of HP performance. The measured performance was used in a correlation model of the HP unit in an annual load-cycle system performance simulation based on the TRNSYS simulation package. The two ASHPWHs tested had significantly lower performance than typical solar WHs or solar-boosted HPWHs. They could be used in applications where solar WHs cannot be considered. These schematic diagrams are given in Figs. 10 and 11.

Zhang et al. [5] dealt with the system optimization of ASHPWH, including calculating and testing (Fig. 12). The ASHPWH system consisted of a HP, a water tank and connecting pipes. Air energy was absorbed at the evaporator and pumped to storage tank via a Rankine cycle. The coil pipe/condenser released condensing heat of the refrigerant to the water side. An ASHPWH using a rotary compressor heated the water from initial temperature to the set temperature (55 °C). The capillary tube length, the filling quantity of refrigerant, the condenser coil tube length and system matching were discussed accordingly.

One other type HP is solar-assisted HP. The oil crisis in the beginning of the 1970s has led researchers to use alternative energy sources for energy production. Solar energy was the first choice. Later, when HPs became popular for heating and cooling applications, combining solar energy and HPs in several ways were

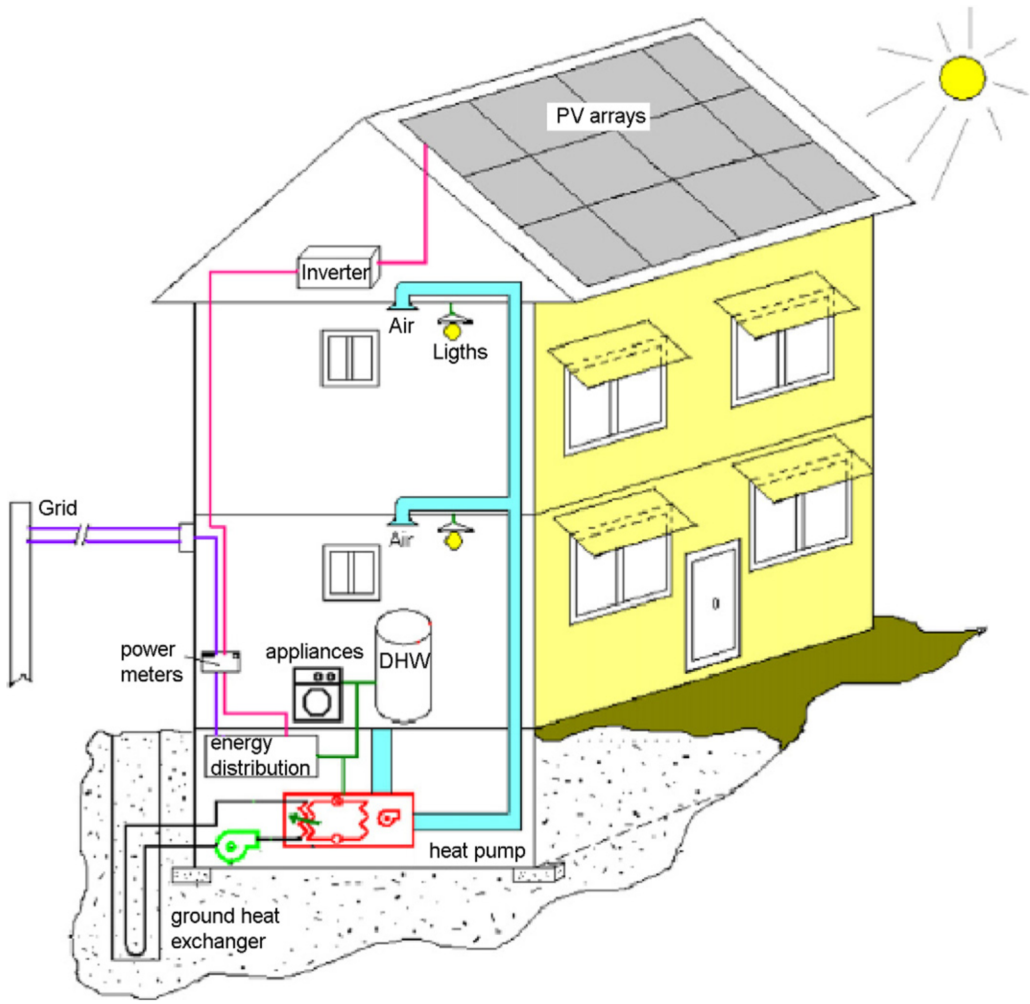

Fig. 4. Schematic representation of the ZNEH [18].

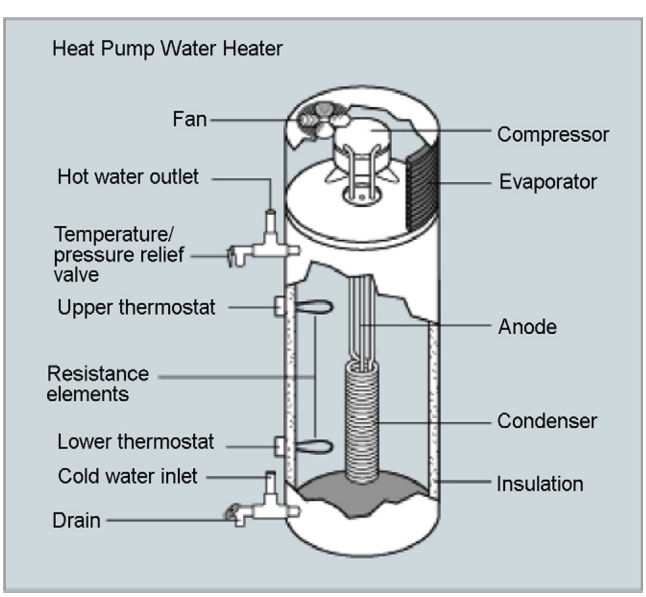

Fig. 5. Cutaway view of a heat pump water heater [2,19].

proposed by many researchers. Although the idea of a DX-SAHP was first proposed by Sporn and Ambrose in 1955, because the first main studies on the subject began at that date, the beginning of the studies on DX-SAHP systems is therefore accepted as the late 1970s [4].

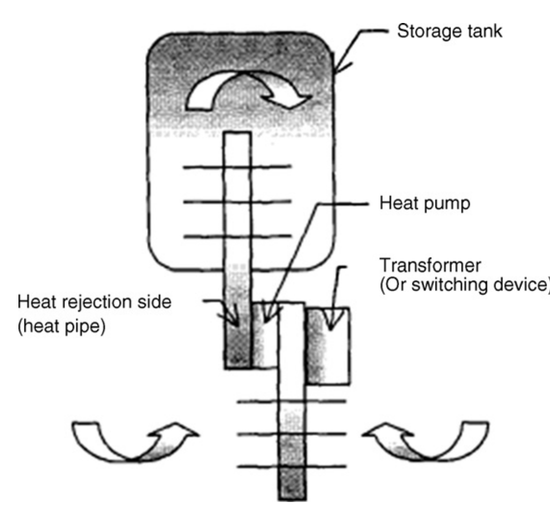

Fig. 6. Heat up stage [20].

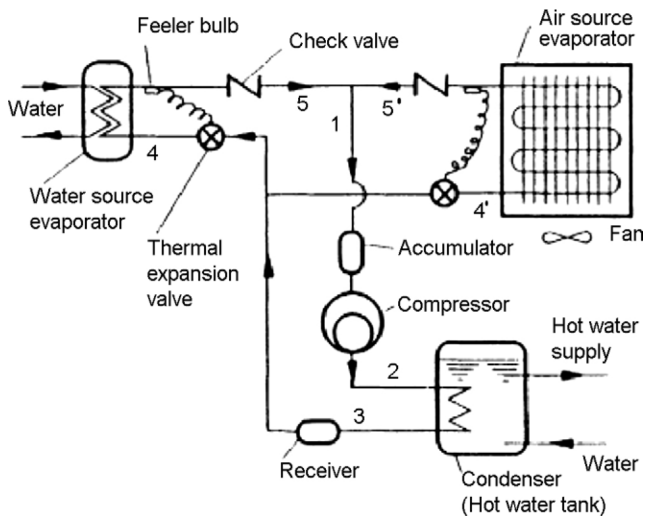

Fig. 7. Heat pump using dual heat sources in parallel arrangement for evaporation [21].

Sakai et al. [24] described a specific HP system that could solve the problem of low heating capacity at a low ambient temperature one of the largest problems in the ASHP system. In order to decrease the collector area required, the HP system was operated by the air-source during the daytime, but at night or at a very low ambient temperature it could be operated with hot water, which has been produced by the collector in the daytime. The effect of the solar energy on the ASHP system had many advantages in the moderate winter climate of Japan. The hot water supply system included an auxiliary electric heater. The experiment had been carried out with a prefabricated test house, which had been constructed in Nara with double glazed windows and high thermal insulation. The results of this experiment were that solar energy enhanced the total electric energy savings, increased the heating capacity at low ambient temperature, and eliminated the need for reverse cycle defrosting operation, etc.

A variable capacity direct-expansion solar-assisted HP system was developed and operated for DHW application by Chaturvedi et al. [25]. The proposed system employed a bare solar collector which also acted as the system evaporator. A variable frequency drive modulated the compressor speed to maintain a proper matching between the heat pumping capacity of the compressor and the evaporative capacity of the collector under widely varying ambient conditions (Fig. 13). Experimental results indicated that the coefficient of performance of the system could be improved significantly by lowering the compressor speed as ambient temperature rises from winter to summer months.

Chaichana et al. [26] presented the comparative assessment of natural working fluids with R-22 in terms of their characteristics and thermo physical properties, and thermal performance. Some justification was given for using natural working fluids in a solar-boosted HPWH. The results showed that R-744 was not suitable for solar-boosted HPs because of its low critical temperature and high operational pressures. On the other hand, R-717 seemed to be a more appropriate substitute in terms of operational parameters and overall performance. R-290 and R-1270 were identified as candidates for direct drop-in substitutes for R-22.

A solar-assisted HP dryer and WH were designed, fabricated and tested by Hawlader et al. [27]. The performance of the system had been investigated under the meteorological conditions of Singapore. A simulation program was developed using Fortran language to evaluate the performance of the system and the influence of different variables (Fig. 14). The performance indices considered to evaluate the performance of the system included solar fraction (SF) and COP with and without a WH. The values of COP, obtained from the simulation and experiment were 7.0, and 5.0, respectively, whereas the SF values of 0.65 and 0.61 were obtained from simulation and experiment, respectively.

Chyng et al. [28] carried out a modeling and system simulation of an integral-type solar-assisted heat pump water heater (ISAHP). The schematic diagram is shown in Fig. 15. The modeling and simulation assumed a quasi-steady process for all the components in the ISAHP except the storage tank. The simulation results for instantaneous performance agreed very well with experiment. The

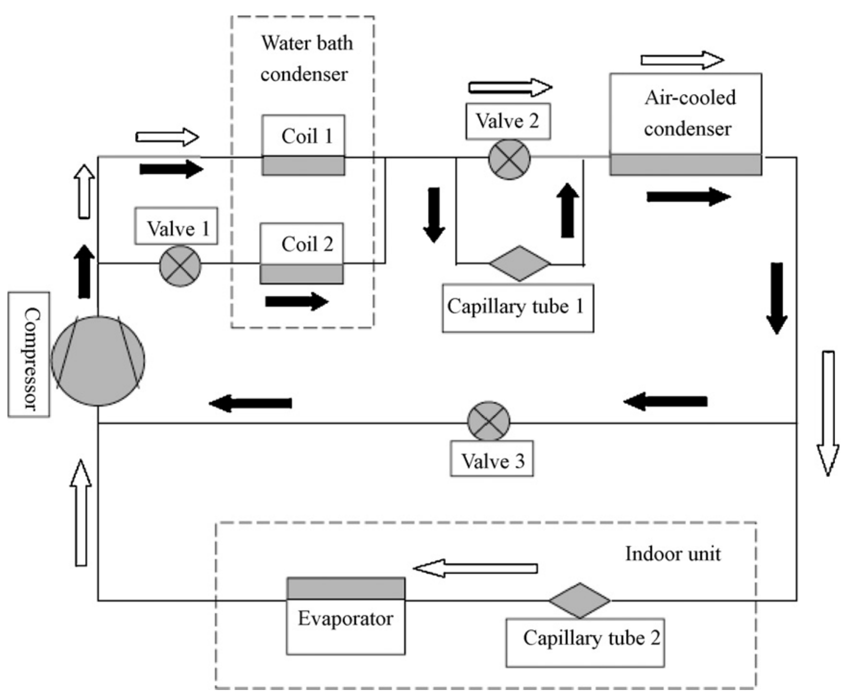

Fig. 8. Working cycles of the two-function air-conditioner [22].

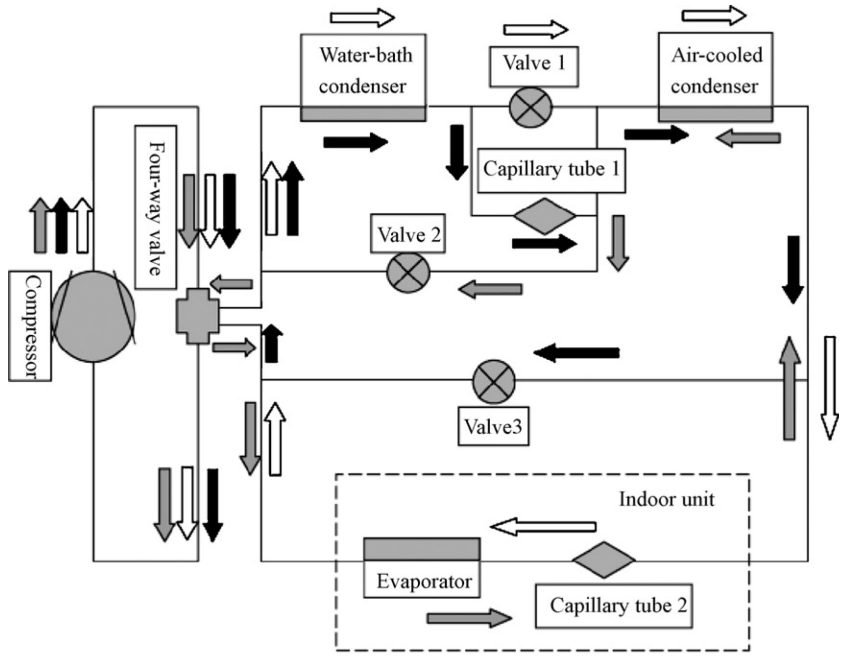

Fig. 9. Working cycles of the three-function air-conditioner [22].

simulation technique was used to analyze the daily performance of an ISAHP for 1 year. It was shown that the daily total COP was around 1.7–2.5 year around for the ISAHP, depending on seasons and weather conditions.

Analytical and experimental studies were performed on a DX-SAHP water heating system, in which a 2 m^2 bare flat collector acted as a source as well as an evaporator for the refrigerant by Kuang et al. [29]. A simulation model was developed to predict the long-term thermal performance of the system approximately. The monthly averaged COP was found to vary between 4 and 6, while the collector efficiency ranged from 40 to 60% (Fig. 16).

Huang et al. [30] studied a heat-pipe enhanced solar-assisted heat pump water heater (HPSAHP). HPSAHP is a heat pump with dual heat sources that combines the performance of conventional heat pump and solar heat-pipe collector. HPSAHP operates in heat-pump mode when solar radiation is low and in heat-pipe mode

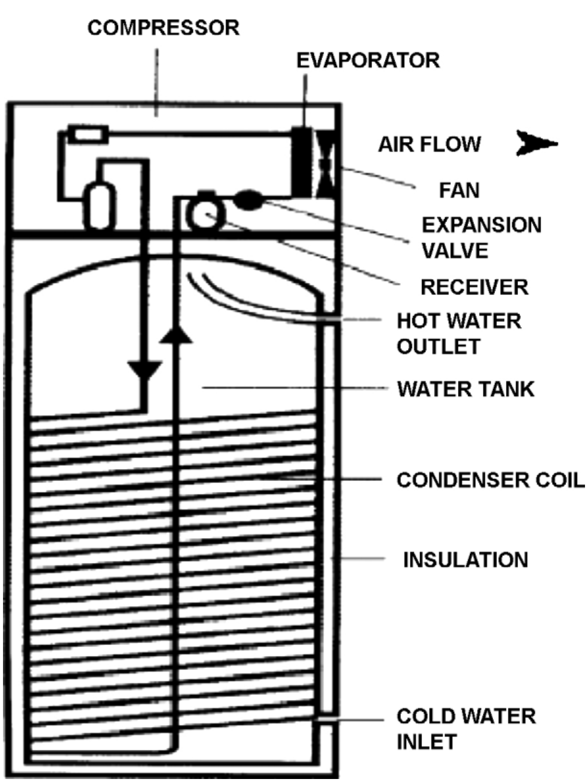

Fig. 10. Air-source heat pump water heater with wrap-around condenser coil [23].

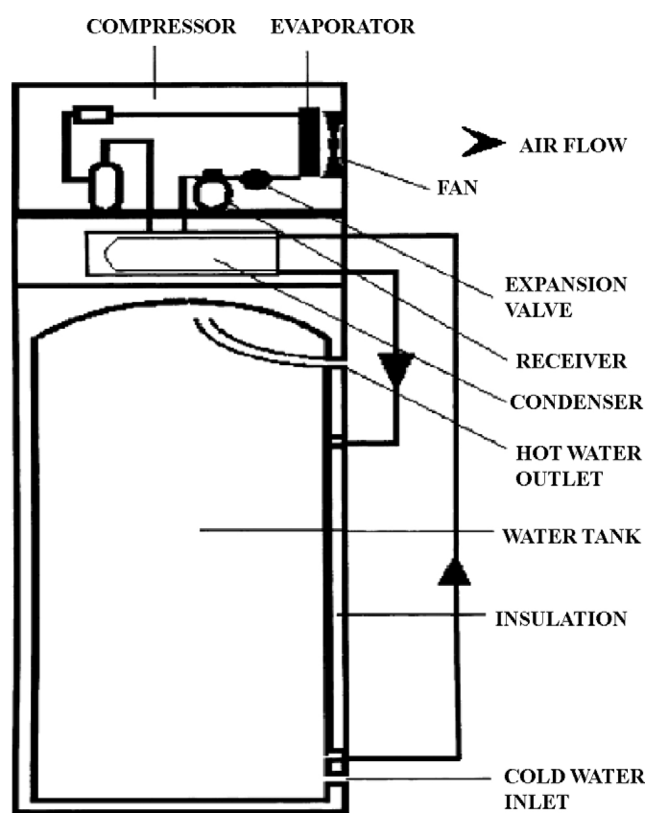

Fig. 11. Air-source heat pump water heater with external condenser [23].

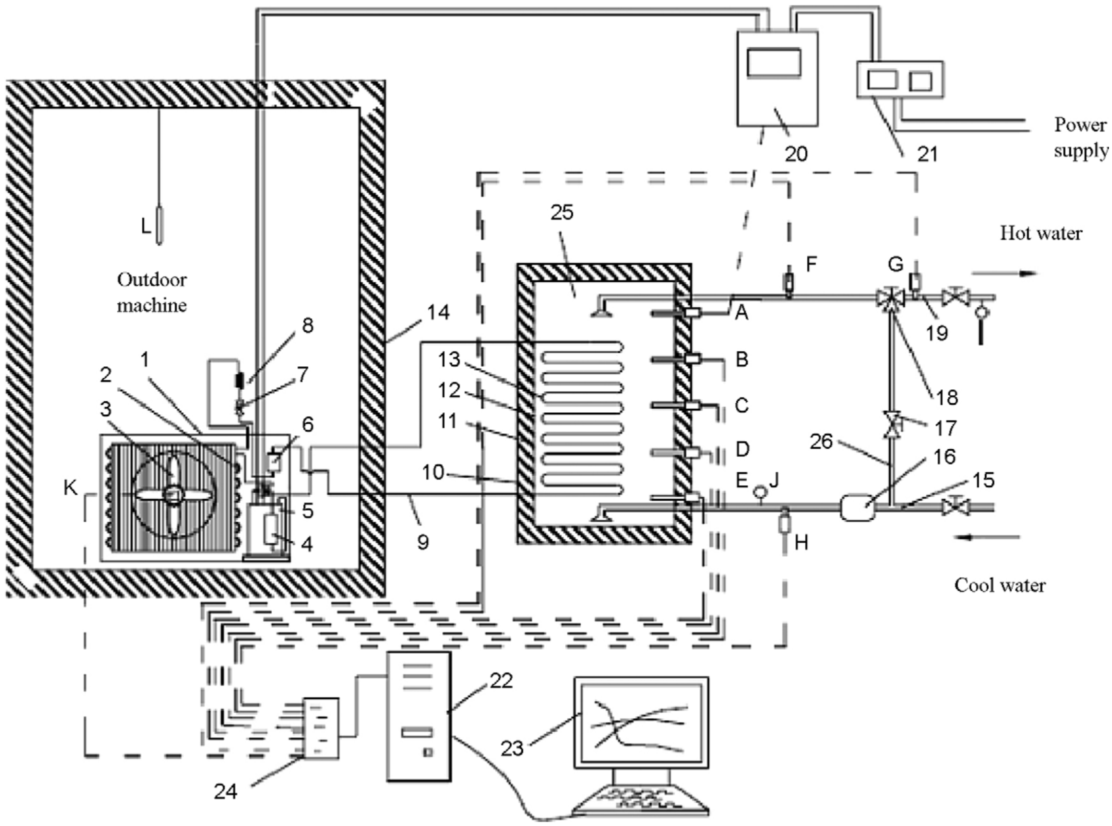

Fig. 12. Sketch of the experimental system to test the performance of ASHPWH: (1) shell body; (2) heat exchanger; (3) fan; (4) store; (5) compressor; (6) filter; (7) valve; (8) thermal expansion valve; (9) copper pipe; (10) outer shell; (11) thermal insulation; (12) inner tank; (13) coil pipe condenser; (14) temperature and humidity controlled room; (15) inlet water pipe; (16) cycle water pump; (17) water mixing valve; (18) three way valve; (19) outlet water pipe; (20) controller; (21) ammeter; (22) computer; (23) data scrutiny; (24) data logger; (25) hot water tank; (26) cycle water pipe; A–H: temperature sensors; I, J: water meters [5].

without electricity consumption when solar radiation is high. HPSAHP can thus achieve high energy efficiency. A prototype was designed and built in the present study. An outdoor test for a HPSAHP in the present study has shown that COP of the hybrid-mode operation can reach 3.32, an increase of 28.7% as compared to the heat-pump mode COP (2.58). The schematic diagram is shown in Fig. 17.

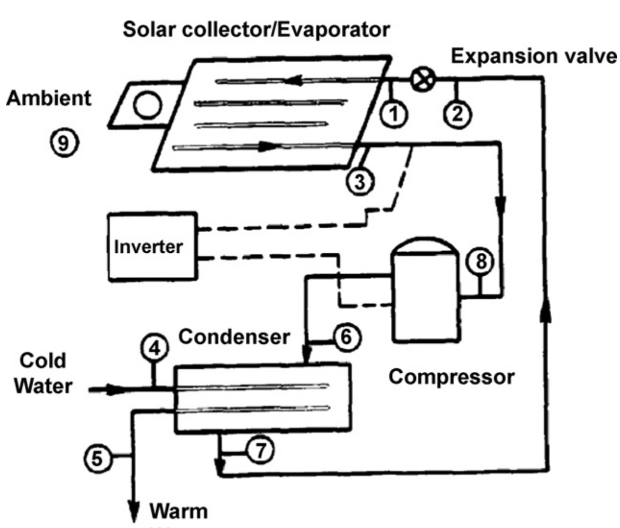

Fig. 13. Schematic of DX-SAHP system [25].

Guoying et al. [31] presented a simulation study on the operating performance of a new type of solar-air source heat pump water heater (SAS-HPWH). The SAS-HPWH used a specially designed flat-plate heat collector/evaporator with spiral-finned tubes to obtain energy from both solar irradiation and ambient air for hot water heating. Using the meteorological data in Nanjing, China, the simulation results based on 150 l water heating capacity showed that such a SAS-HPWH could heat water up to 55 °C efficiently under various weather conditions all year around. This schematic diagram is illustrated in Fig. 18.

One another study (Fig. 19) on DX-SAHP was conducted by Li et al. [32]. A direct-expansion solar-assisted heat pump water heater (DX-SAHPWH) experimental set-up was introduced and analyzed. This DX-SAHPWH system mainly consisted of 4.20 m^2 direct-expansion type collector/evaporator, R-22 rotary-type hermetic compressor with rated input power 0.75 kW, 150-l water tank with immersed 60-m serpentine copper coil and external balance type thermostatic expansion valve. The experimental research under typical spring climate in Shanghai showed that the COP of the DX-SAHPWH system could reach 6.61 when the average temperature of 150 l water was heated from 13.4 to 50.5 °C in 94 min with average ambient temperature 20.6 °C and average solar radiation intensity 955 W/m^2. The COP of the DX-SAHPWH system was 3.11 even if at a rainy night with average ambient temperature 17.1 °C. The seasonal average value of the COP and the collector efficiency were measured to be 5.25 and 1.08, respectively.

Hepbasli [33] studied on the exergetic modeling and performance evaluation of solar-assisted DHW tank integrated GSHP

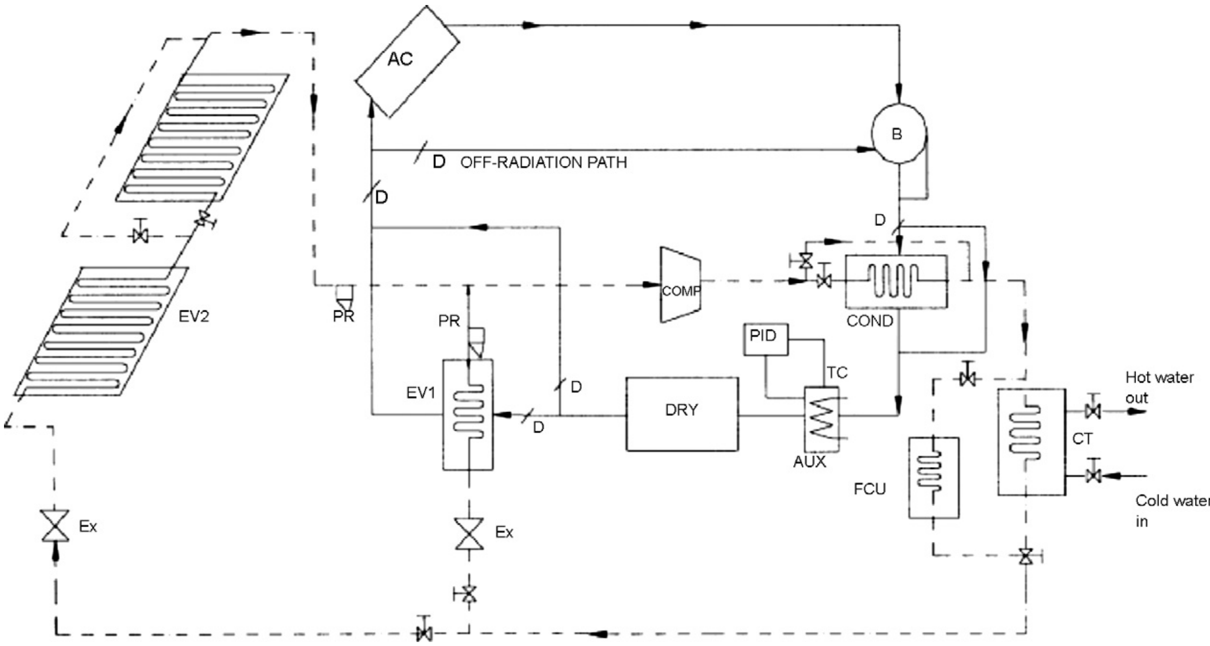

Fig. 14. Schematic diagram of heat-pump assisted solar-dryer and water heater: (AUX) auxiliary heater; (AC) air collector; (B) blower; (COMP) compressor; (COND) condenser; (CT) condenser tank; (D) damper; (DRY) dryer; (EV1) evaporator 1; (EV2) evaporator 2; (EX) expansion valve; (TC) thermocouple; (PID) temperature controller; (-) air path; (- -) refrigerant path; (PR) pressure regulator; (FCU) fan-coil unit [27].

systems for residences (Fig. 20), while he used the existing energetic data given by Trillat-Berdal et al. [34]. Exergy efficiency values on a product/fuel basis were found to be 72.33% for the GSHP unit, 14.53% for the solar DHW system and 44.06% for the whole system at dead (reference) state values for 19 °C and 101.325 kPa. Exergetic COP values were obtained to be 0.245 and 0.201 for the GSHP unit and the whole system, respectively. The greatest irreversibility (exergy destruction) on the GSHP unit basis occurred in the condenser, followed by the compressor, expansion valve and evaporator.

One other type HPs are new technology GEHPs. Although GEHP was appeared on the Japanese HVAC marketplace in 1986 and

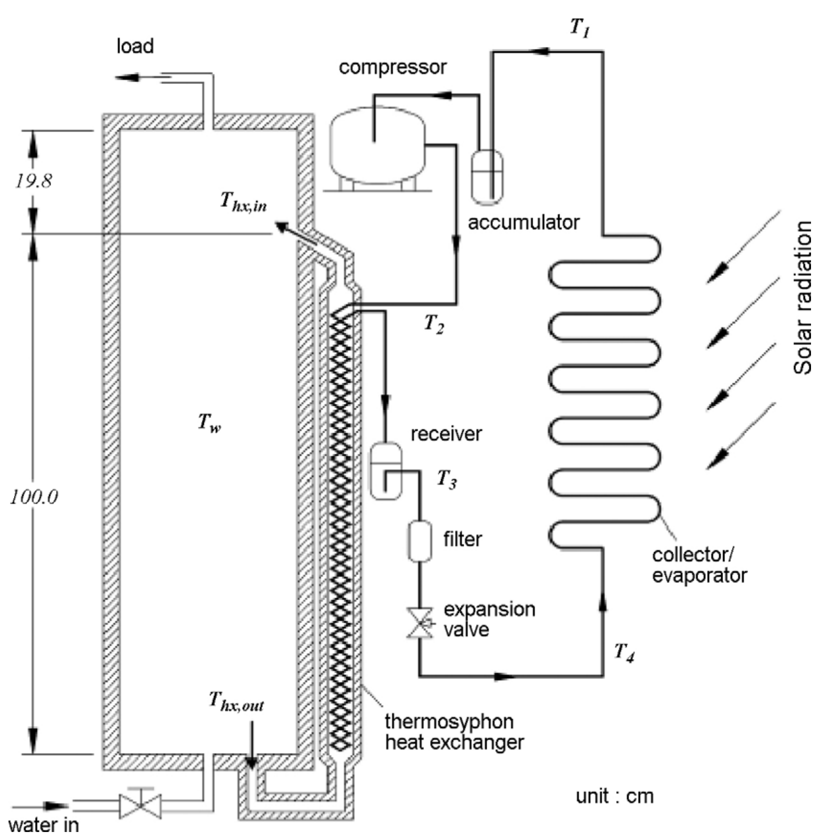

Fig. 15. Schematic diagram of the integral-type solar-assisted heat pump [28].

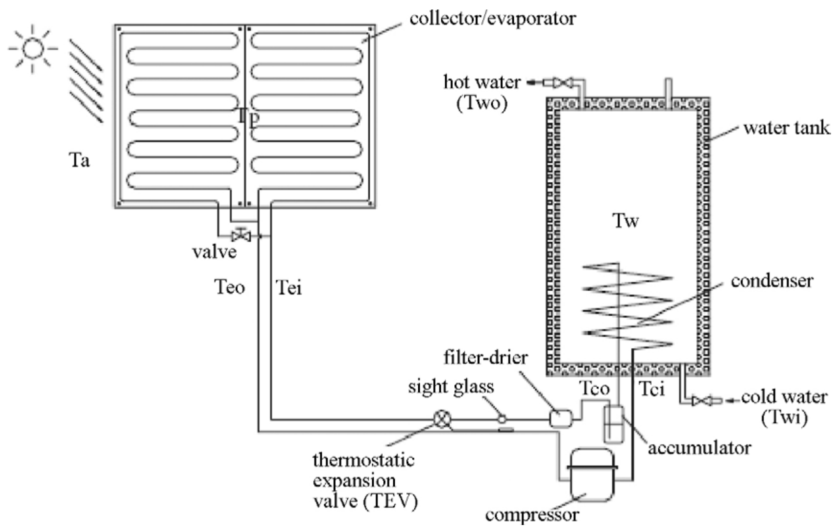

Fig. 16. Schematic diagram of the DX-SAHP water heater [29].

water-loop heat pump system (WLHPS) even as early as 1960s in the USA, the combination of GEHP and WLHPS is completely a new conception of their application [35].

A GEHP usually consists of a reversible vapor compression HP with an open compressor driven by a gas fueled internal combustion engine instead of an electric motor. The GEHP has been paid more attention in the heating, ventilation and air conditioning (HVAC) field in recent years due to its advantage of reducing the electric consumption in the cooling and heating seasons. Another two distinguished advantages of the GEHP are (1) the ability to recover the waste heat released by the engine cylinder jacket and exhaust gas in the heating mode and (2) the easy modulation of compressor speed by adjusting the gas supply. Therefore, the GEHP has a different performance from that of the electric driven heat pump (EHP), especially in the heating mode [36].

Conception of combination of GEHP and WLHPS was firstly presented by Lian et al. [35] in order to reduce the energy consumption of air conditioning system further. Design of the new system was introduced through an actual project in China and

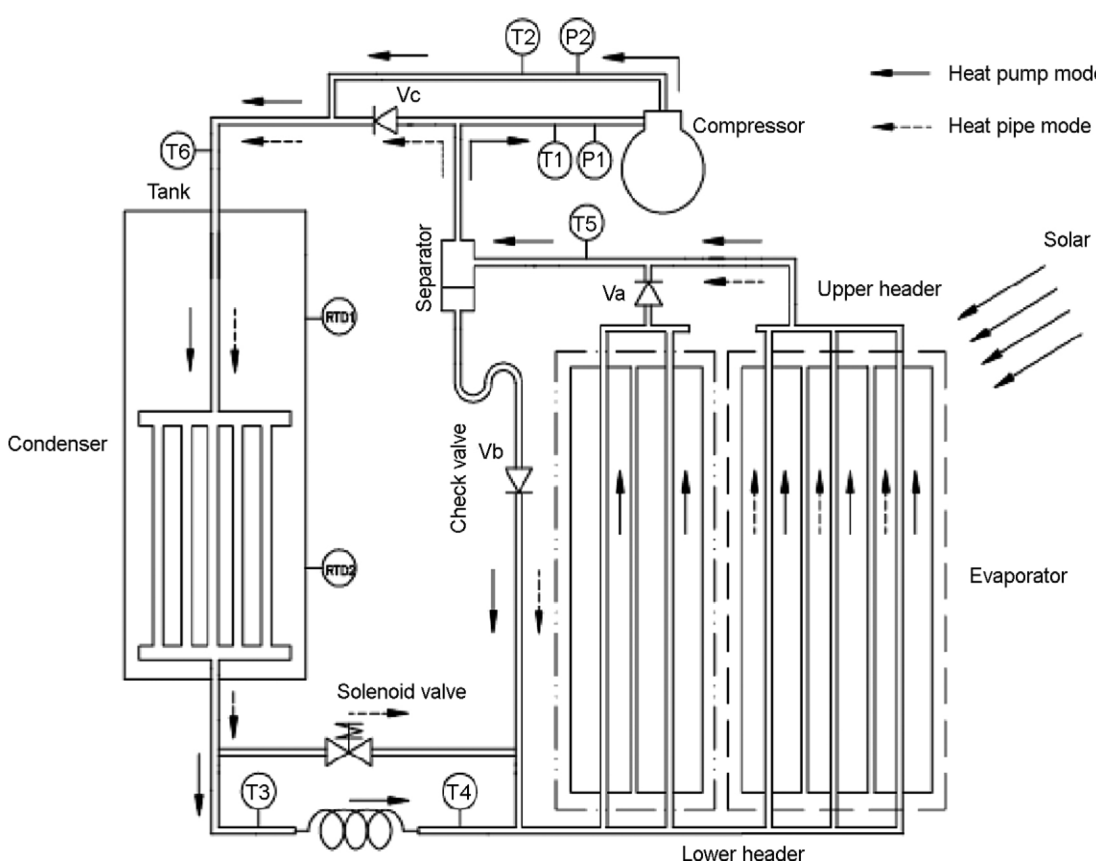

Fig. 17. Flow chart and schematic diagram of the HASHP [30].

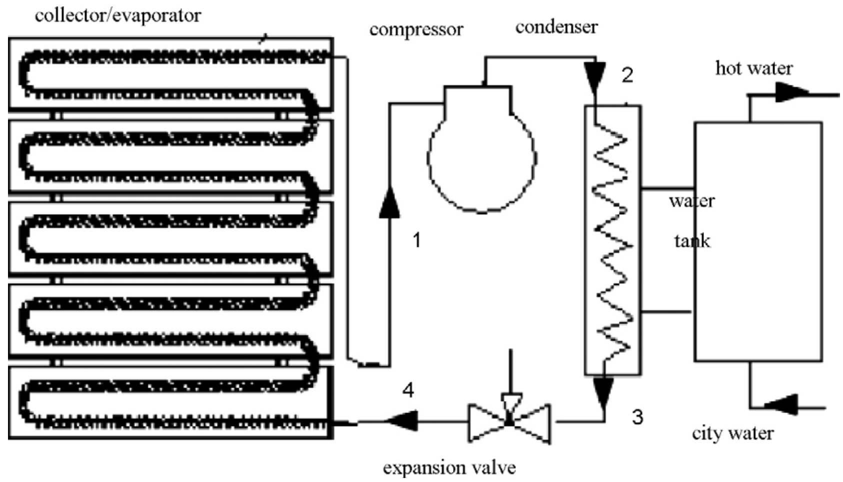

Fig. 18. The schematic diagram of the simulated SAS-HPWH [31].

compared with a conventional air-conditioning system (CACS) and conventional WLHPS (EHP-WLHPS) in terms of technical characteristics and payback period. It was found that the payback period of GEHP-WLHPS was about 2 years when compared with CACS and 2.6 years with EHP-LHPS on the average. The schematic diagram is shown in Fig. 21.

Zhang et al. [36] analyzed the heating performance of a gas engine driven air to water HP using a steady state model (Fig. 22). The thermodynamic model of a natural gas engine was identified by the experimental data, while the compressor model was created by several empirical equations. The heat exchanger models were developed by the theory of heat balance. The system model was validated by comparing the experimental and simulation data, which indicated good agreement. The results showed that engine waste heat could provide about 1/3 of the total heating capacity in this gas engine driven air to water HP. The performance of the engine, HP and integral system were analyzed under variations of engine speed and ambient temperature. It indicated that engine speed had remarkable effects on both the engine and HP, but ambient temperature had little influence on the engine's performance.

Lazzarin and Noro [37] analyzed "S. Nicola" HVAC plant in Vicenza features and significant energy savings characteristics (Fig. 23). It has been set up by a GEHP (coupled to two condensing boilers) whose performances were evaluated during 3 years of operation. In the following analysis only the heating and cooling

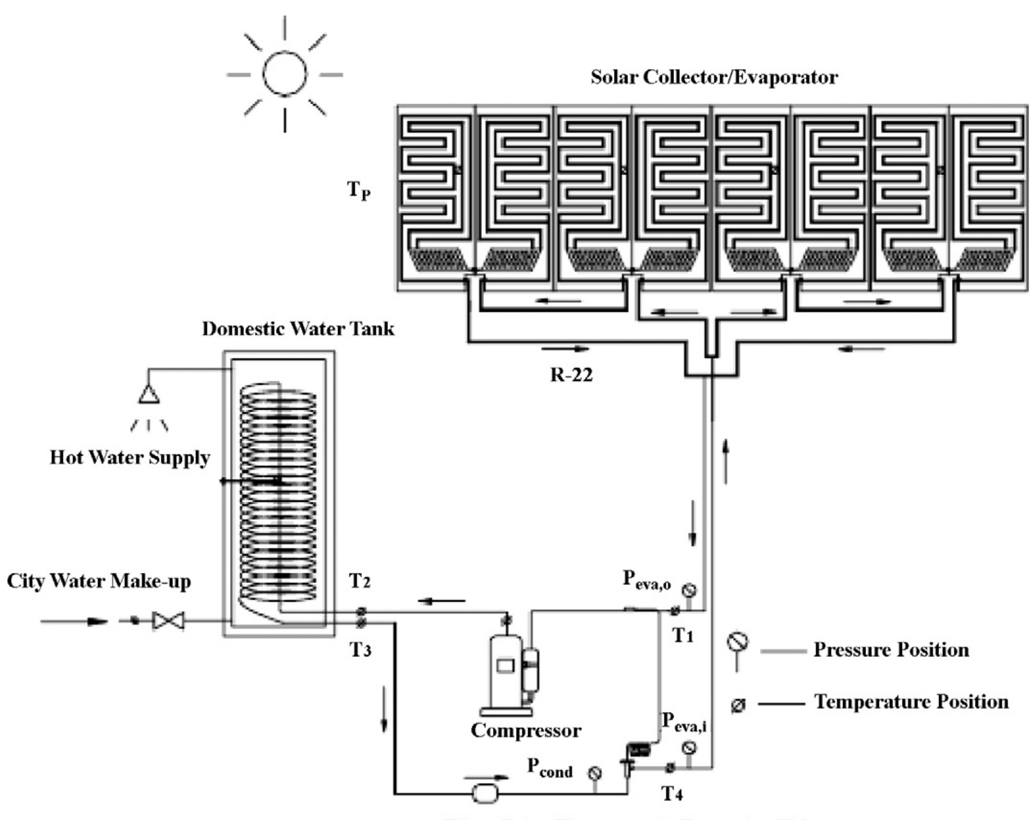

Fig. 19. Schematic of system circuit [32].

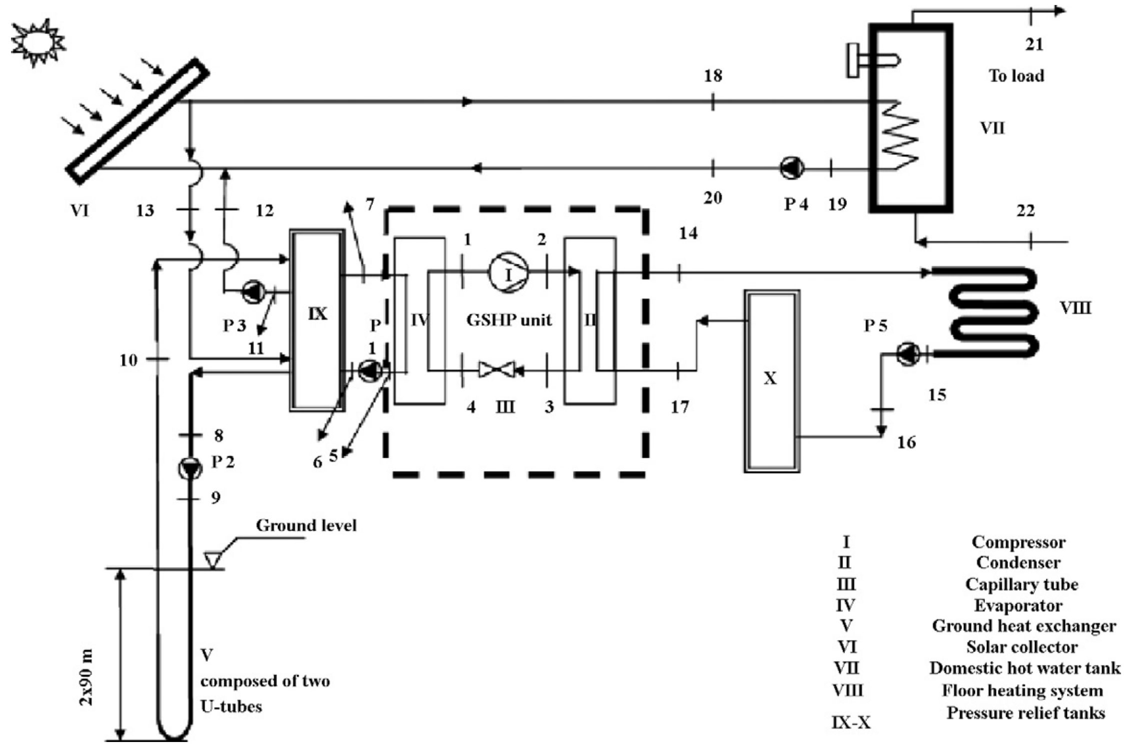

Fig. 20. Schematic view of a solar-assisted domestic hot water tank integrated GSHP system [33,34].

plant had been considered. The main component was reversible air/water GEHP. The nominal cooling power was 275 kW (22 Nm^3/h of natural gas was the nominal fuel consumption), while in heating mode the output power was 380 kW (19 Nm^3/h fuel consumption). These performances were labeled for summer external air 35 °C and evaporator input/output 12/7 °C; winter external air 10 °C and condenser input/output 40/45 °C. Heat recovery was taken from the lubricating oil, engine cooling water and partly from the exhaust. The nominal power thus recovered was 109 kW in heating mode and 127 kW in cooling mode, to produce hot water at about 70 °C.

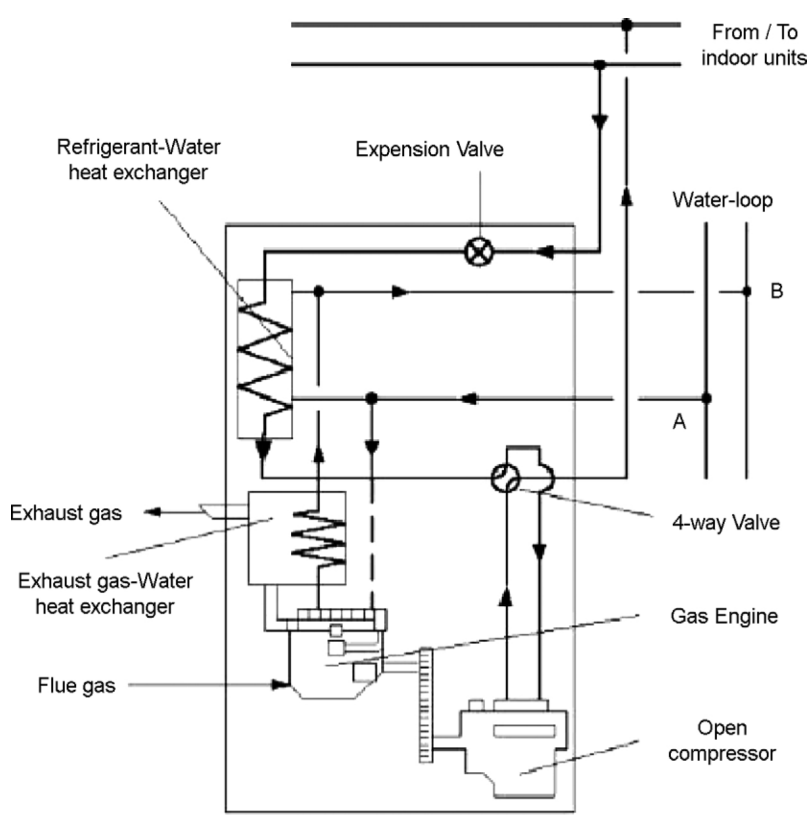

Fig. 21. Schematic view of the outdoor unit of a GEHP [35].

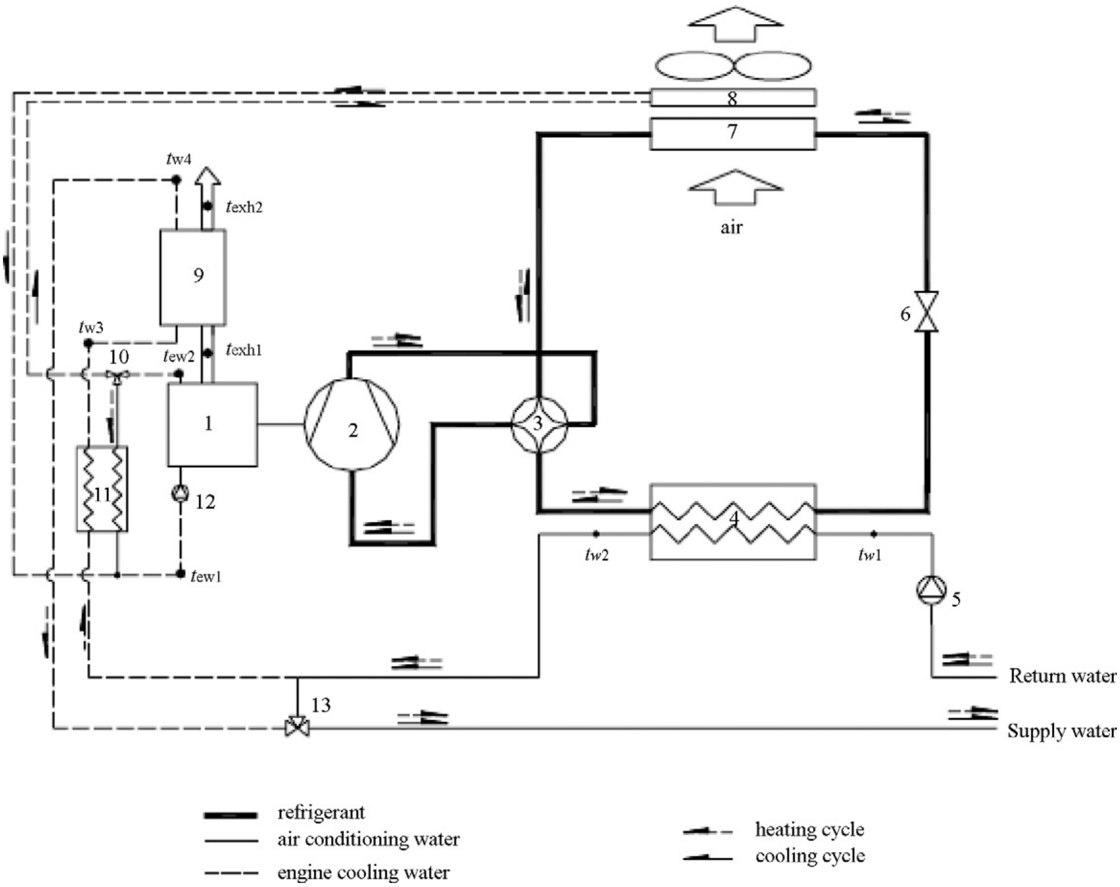

Fig. 22. Schematic diagram of GEHP: (1) natural gas engine, (2) open compressor, (3) four-way valve, (4) plate heat exchanger, (5) supply water pump, (6) expansion valve, (7) finned-tube heat exchanger, (8) heat radiator, (9) gas-to-water heat exchanger, (10) three-way valve, (11) water-to-water heat exchanger, (12) cooling water pump, (13) three-way valve [36].

Historical development of GEHP systems was briefly given by Hepbasli et al. [38]. Next, the operation of these systems was described, while studies conducted on them were reviewed and presented in tabulated forms. GEHPs were then modeled for performance evaluation purposes by using energy and exergy analysis methods. Finally, an illustrative example was given.

4.2. Classifying studies conducted

A detailed review of studies conducted on HPWH systems is given in Table 1. As can be seen from the table, HPWH systems were evaluated under four groups, GSHP, ASHP, solar-assisted HP and GEHP, respectively.

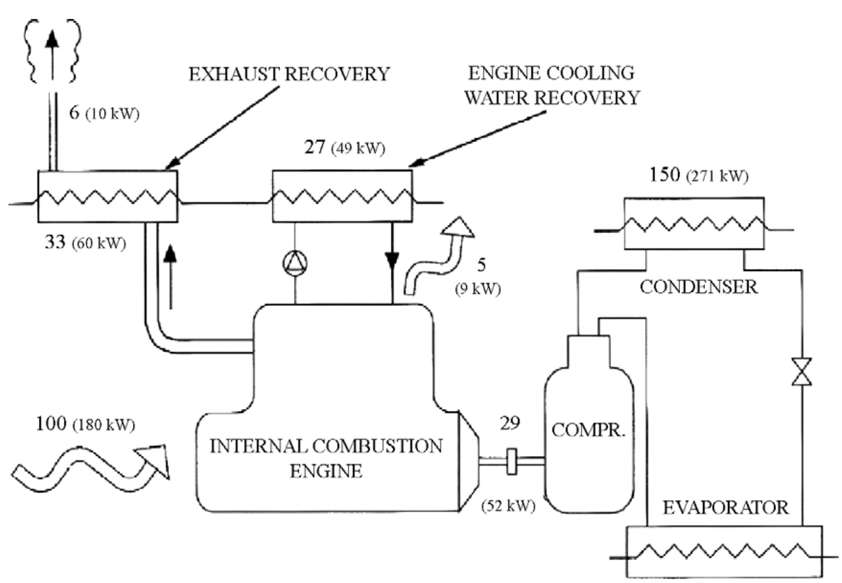

Fig. 23. Schematic view of the gas engine heat pump installed in the "S. Nicola" HVAC plant (values reported refer to heating mode) [37].

Table 1
Main studies conducted on HPWH systems

Classification	#	Year	Investigator(s)	Type of study		Type of system		Application type			Type of analyze		Results
				Theoretical (simulation)	Experimental	Heating	Cooling	Water heating	A/C	Other	Energy	Exergy	
Ground-source heat pump	1	2001	Swardt and Meyer [16]	√	√	√	√	√	√		√		For simulation/experimental: $COP_{heating\text{-}air\ source} = 3.315/2.998$; $COP_{cooling\text{-}air\ source} = 2.743/2.504$; $COP_{heating\text{-}ground\ source} = 3.320/3.307$; $COP_{cooling\text{-}ground\ source} = 3.190/3.049$
	2	2003	Hepbasli et al. [17]		√	√		√	√		√		$COP_{heating} = 1.656$; $COP_{system} = 1.339$
	3	2008	Biaou and Beriner [18]	√		√	√	√	√		√		Heat pump water heater: COP = 2.4
Air-source heat pump	4	1979	Sloane et al. [19]	√	√	√		√			√		COP = about 2.0–2.8 (18–35 °C ambient air, 27 and 4.4 °C source water)
	5	1998	Harata et al. [20]		√	√		√			√		Thermoelectric technology storage tank water heaters about a 13% reduction in energy consumption were achieved
	6	2000	Ito and Miura [21]	√	√	√		√			√		$COP = f(T_e) T_e$: evaporation temperature. Run 1 at an air temperature of 20 °C; COP = 4.0. Run 2 the temperature of the water and the air 10 °C; COP = 3.68
	7	2003	Ji et al. [22]		√	√	√	√	√		√		For three function model—Mode 1: space-cooling and water heating; $COP_{cw\text{-}avg} = 4.02$, $COP_{c\text{-}avg} = 2.91$. Mode 2: water heating only; $COP_{w\text{-}avg} = 3.42, 3.25, 2.52, 2.00$ (for $T_0 = 31, 25, 15, 4.5$ °C). Mode 3: space heating only $COP_h = 2.72$
	8	2004	Morrison et al. [23]	√	√	√		√			√		In Sydney Australia 40 Mj/day peak winter load: $COP_{integral\text{-}condenser} = 2.3$ and annual energy saving = 56%; $COP_{external\text{-}condenser} = 1.8$ and annual energy saving = 44%
	9	2007	Zhang et al. [5]		√	√		√			√		$COP_{winter} = 2.61$ (for $T_0 = 0$ °C); $COP_{summer} = 5.66$ (for $T_0 = 35$ °C); $COP_{spring/autumn} = 4.817$ (for $T_0 = 25$ °C)
Solar-assisted heat pump	10	1976	Sakai et al. [24]		√	√	√	√	√		√		–
	11	1996	Chaturvedi et al. [25]	√	√	√		√			√		(in the 30–70 Hz frequency range): $COP_h = 2.5–4.0$
	12	2003	Hawlader et al. [27]	√	√	√		√			√		$COP_{system} = 6.0$; $\eta_{evap\text{-}coll} = 0.080$, $\eta_{air\text{-}coll} = 0.77$
	13	2003	Chyng et al. [28]	√	√	√		√			√		$COP_{daily\text{-}total} = 1.7–2.5$ (year around); $T_{water} = 57.2$ °C
	14	2003	Kuang et al. [29]	√	√	√		√			√		$COP_{monthly\text{-}avg} = 4–6$; $\eta_{coll} = 40–60\%$
	15	2005	Huang et al. [30]		√	√		√			√		$COP_{HP\text{-}mode} = 2.58$; $COP_{hybrid\text{-}mode} = 3.32$
	16	2006	Guoying et al. [31]	√		√		√			√		The monthly averaged COP = 3.98–4.32; $T_{water} = 55$ °C
	17	2007	Li et al. [32]		√	√		√			√	√	$COP_{seasonal\text{-}avg} = 5.25$; $\eta_{coll} = 1.08$; $\varepsilon_{system} = 21\%$; $T_{water} = 50.5$ °C
	18	2007	Hepbasli [33,34]		√	√		√	√		√	√	$\varepsilon_{GSHP} = 72.33\%$, $\varepsilon_{SDHWS} = 14.53\%$, $\varepsilon_{SYSTEM} = 44.06\%$. Exergetic COP: $COP_{GSHP} = 0.245$, $COP_{SYSTEM} = 0.201$
Gas engine-driven heat pump	19	2005	Lian et al. [35]		√	√	√	√	√		√		Payback period years: GEHP-WLHPS 2; EHP-WLHPS 2.6
	20	2005	Zhang et al. [36]	√	√	√	√	√	√		√		$\eta_{thermal} = 28.6\%$ at max. output; $\eta_{engine} = 29.7\%$ for $T_0 = 5$ °C
	21	2006	Lazzarian and Noro [37]	√	√	√	√	√	√		Economically evaluated		$Q_{cooling} = 275$ kW, $Q_{heating} = 380$ kW, $T_{water} = 70$ °C. For 2001 data district heating thermal unit price about = 5.5 c€/kW h_t
	22	–	Hepbasli et al. [38]	√		√	√	√	√				A detailed review on GEHPs was made

Table 2
Comparison of water heaters [39]

High efficiency water heater type	Energy savings vs. minimum standards	Best climates	Expected energy savings over equipment lifetime (US$)	Expected lifetime (years)	Major advantages
High efficiency storage (tank) (oil, gas, elec.)	10–20%	Any	Up to 500	8–10	Lowest first cost
Demand (tankless) using gas or elec.	45–60%	Any	Up to 1800	20	Unlimited supply of hot water
Heat pump	65% (compared to electric resistance)	Mild-hot	Up to 900	10	Most efficient electric fuel option
Solar with electric back-up	70–90%	Mild-hot	Up to 2200	20	Largest energy savings using a renewable energy source

The performance indicator for HP systems, namely COP values, varied from 0.201 to 6, while most of the values were between 2 and 3.

Most of the systems were used in water-heating applications, space heating and A/C performances. Since energy was mainly used for water and space-heating purposes in many countries, studies related with improving the performances and reducing costs in such applications played an important role, also providing less demand for fossil fuels. Theoretical and experimental analysis as well as energy analysis were conducted by nearly all of the studies, while exergy analysis was covered by very few of them.

Comparison of WHs is seen in Table 2. According to the table, the most efficiency WH types are HP and solar with electric back-up with energy savings vs. minimum standards, 65% and 70–90%, respectively. Expected life time for these systems are 10 and 20 years, while major advantages are most efficient electric fuel option and largest energy savings using a renewable energy source, respectively [39].

5. Modeling

Energy and exergy analysis is presented for the performance evaluation of the HPWH systems. Balance (mass, energy and exergy) equations for steady state, constant-flow control volume systems, and appropriate energy and exergy equations are derived for these systems and its components [32,40,41].

5.1. General energy and exergy balance equations

The mass balance equation can be expressed in the rate form as

$$\sum \dot{m}_{\mathrm{in}} = \sum \dot{m}_{\mathrm{out}} \tag{1}$$

where $\dot{m}$ is the mass flow rate, and the subscript in stands for inlet and out for outlet.

The general energy and exergy balances can be expressed below as the total energy and exegy inputs equal to total energy and exergy outputs, respectively.

$$\sum \dot{E}_{\mathrm{in}} = \sum \dot{E}_{\mathrm{out}} \tag{2}$$

$$\sum \dot{E}x_{\mathrm{in}} - \sum \dot{E}x_{\mathrm{out}} = \sum \dot{E}x_{\mathrm{dest}} \tag{3a}$$

or

$$\dot{E}x_{\mathrm{heat}} - \dot{E}x_{\mathrm{work}} + \dot{E}x_{\mathrm{mass,in}} - \dot{E}x_{\mathrm{mass,out}} = \dot{E}x_{\mathrm{dest}} \tag{3b}$$

Using Eq. (3b), the rate form of the general exergy balance can also be written as

$$\sum \left(1 - \frac{T_0}{T_{\mathrm{k}}}\right)\dot{Q}_{\mathrm{k}} - \dot{W} + \sum \dot{m}_{\mathrm{in}}\psi_{\mathrm{in}} - \sum \dot{m}_{\mathrm{out}}\psi_{\mathrm{out}} = \dot{E}x_{\mathrm{dest}} \tag{4}$$

with

$$\psi = (h - h_0) - T_0(s - s_0) \tag{5}$$

where $\dot{Q}_{\mathrm{k}}$ is the heat transfer rate through the boundary at temperature T_{k} at location k, $\dot{W}$ is the work rate, ψ is the flow (specific) exergy, h is enthalpy, s is entropy, and the subscript zero indicates properties at the restricted dead state of P_0 and T_0.

Multiplying flow or specific exergy given in Eq. (5) by the mass flow rate of the fluid gives the exergy rate:

$$\dot{E}x = \dot{m}[(h - h_0) - T_0(s - s_0)] \tag{6}$$

Also, it is usually more convenient to find $\dot{S}_{\mathrm{gen}}$ first and then to evaluate the exergy destroyed or the irreversibility rate $\dot{I}$ directly from the following equation, which is called Gouy–Stodola relation:

$$\dot{I} = \dot{E}x_{\mathrm{dest}} = T_0\dot{S}_{\mathrm{gen}} \tag{7}$$

Furthermore, mass, energy and exergy equations are demonstrated in Table 3 for each components.

5.2. Improvement potential

Van Gool [42] has also proposed that maximum improvement in the exergy efficiency for a process or system is obviously achieved when the exergy loss or irreversibility $(\dot{E}x_{\mathrm{in}} - \dot{E}x_{\mathrm{out}})$ is minimized. Consequently, he suggested that it is useful to employ the concept of an exergetic "*improvement potential*" when analyzing different processes or sectors of the economy. This improvement potential in the rate form, denoted $I\dot{P}$, is given by

$$I\dot{P} = (1 - \varepsilon)(\dot{E}x_{\mathrm{in}} - \dot{E}x_{\mathrm{out}}) \tag{8}$$

5.3. Some thermodynamic parameters

Thermodynamics analysis of thermal and HPWH systems may also be performed using the following parameters [43]:

- Fuel depletion ratio:

$$\delta_i = \frac{\dot{I}_i}{\dot{F}_{\mathrm{T}}} \tag{9}$$

- Relative irreversibility:

$$\chi_i = \frac{\dot{I}_i}{\dot{I}_{\mathrm{T}}} \tag{10}$$

Table 3
Balance equations for the system components

Component	Mass analysis using Eq. (1) (the conversion of mass principle)	Energy analysis using Eq. (2) (the first law of thermodynamics or the conversion of energy)	Exergy analysis (the second law of thermodynamics)	
			The irreversibility or the exergy destroyed using Eq. (6) (the exergy balance)	Exergy efficiency $\varepsilon = \dot{P}/\dot{F}$
Compressor (I)	$\dot{m}_{\mathrm{comp,in,r}} = \dot{m}_{\mathrm{comp,out,r}} = \dot{m}_{\mathrm{r}}$	$\dot{W}_{\mathrm{comp}} = \dot{m}_{\mathrm{r}}(h_{\mathrm{comp,out,act}} - h_{\mathrm{comp,in}})$ $\dot{W}_{\mathrm{comp,elec}} = \dot{W}_{\mathrm{comp}}/(\eta_{\mathrm{comp,elec}}\eta_{\mathrm{comp,mech}})$ $\dot{W}_{\mathrm{comp,elec}} = \sqrt{3}V_{\mathrm{comp}}I_{\mathrm{comp}}\cos\varphi$	$\dot{I}_{\mathrm{comp}} = \dot{m}_{\mathrm{r}}(\psi_{\mathrm{comp,in}} - \psi_{\mathrm{comp,out,act}}) + \dot{W}_{\mathrm{comp}}$	$\varepsilon_{\mathrm{comp}} = \frac{\dot{E}x_{\mathrm{comp,out,act}} - \dot{E}x_{\mathrm{comp,in}}}{\dot{W}_{\mathrm{comp,elec}}}$
Condenser (II)	$\dot{m}_{\mathrm{cond,in,r}} = \dot{m}_{\mathrm{cond,out,r}} = \dot{m}_{\mathrm{r}}$ $\dot{m}_{\mathrm{cond,in,w}} = \dot{m}_{\mathrm{cond,out,w}} = \dot{m}_{\mathrm{w}}$	$\dot{Q}_{\mathrm{cond}} = \dot{m}_{\mathrm{r}}(h_{\mathrm{cond,in,act}} - h_{\mathrm{cond,out}})$ $\dot{Q}_{\mathrm{cond}} = \dot{Q}_{\mathrm{fc}} = \dot{m}_{\mathrm{w}}C_{p,\mathrm{w}}(T_{\mathrm{fc,in}} - T_{\mathrm{fc,out}})$ $\dot{Q}_{\mathrm{sh}} = \dot{m}_{\mathrm{air}}C_{p,\mathrm{air}}(T_{\mathrm{out,air}} - T_{\mathrm{in,air}})$	$\dot{I}_{\mathrm{cond}} = \dot{m}_{\mathrm{r}}(\psi_{\mathrm{cond,in,r}} - \psi_{\mathrm{cond,out,r}}) + \dot{m}_{\mathrm{w,fh}}(\psi_{\mathrm{cond,out,w}} - \psi_{\mathrm{cond,in,w}})$	$\varepsilon_{\mathrm{cond}} = \frac{\dot{E}x_{\mathrm{cond,out,w}} - \dot{E}x_{\mathrm{cond,in,w}}}{\dot{E}x_{\mathrm{cond,in,act,r}} - \dot{E}x_{\mathrm{cond,out,r}}} = \frac{\dot{m}_{\mathrm{w,fh}}(\psi_{\mathrm{cond,out,w}} - \psi_{\mathrm{cond,in,w}})}{\dot{m}_{\mathrm{r}}(\psi_{\mathrm{cond,in,act,r}} - \psi_{\mathrm{cond,out,r}})}$
Expansion (throttling) valve (III)	$\dot{m}_{\mathrm{exp,in}} = \dot{m}_{\mathrm{exp,out}} = \dot{m}_{\mathrm{r}}$	$h_{\mathrm{exp,in}} = h_{\mathrm{exp,out}}$	$\dot{I}_{\mathrm{exp}} = \dot{m}_{\mathrm{r}}(\psi_{\mathrm{exp,in}} - \psi_{\mathrm{exp,out}})$	$\varepsilon_{\mathrm{exp}} = \frac{\dot{E}x_{\mathrm{exp,out}}}{\dot{E}x_{\mathrm{exp,in}}} = \frac{\psi_{\mathrm{exp,out}}}{\psi_{\mathrm{exp,in}}}$
Evaporator (IV)	$\dot{m}_{\mathrm{evap,in,r}} = \dot{m}_{\mathrm{evap,out,r}} = \dot{m}_{\mathrm{r}}$ $\dot{m}_{\mathrm{evap,in,w}} = \dot{m}_{\mathrm{evap,out,w}} = \dot{m}_{\mathrm{w}}$	$\dot{Q}_{\mathrm{evap}} = \dot{m}_{\mathrm{r}}(h_{\mathrm{evap,out}} - h_{\mathrm{evap,in}})$ $\dot{Q}_{\mathrm{evap}} = \dot{m}_{\mathrm{w}}c_{p,\mathrm{w}}(T_{\mathrm{evap,in,w}} - T_{\mathrm{evap,out,w}})$	$\dot{I}_{\mathrm{evap}} = \dot{m}_{\mathrm{r}}(\psi_{\mathrm{evap,in,r}} - \psi_{\mathrm{evap,out,r}}) + \dot{m}_{\mathrm{w}}(\psi_{\mathrm{evap,in,w}} - \psi_{\mathrm{evap,out,w}})$	$\varepsilon_{\mathrm{evap}} = \frac{\dot{E}x_{\mathrm{evap,out,w}} - \dot{E}x_{\mathrm{evap,in,w}}}{\dot{E}x_{\mathrm{evap,in,r}} - \dot{E}x_{\mathrm{evap,out,r}}} = \frac{\dot{m}_{\mathrm{w}}(\psi_{\mathrm{evap,out,w}} - \psi_{\mathrm{evap,in,w}})}{\dot{m}_{\mathrm{r}}(\psi_{\mathrm{evap,in,r}} - \psi_{\mathrm{evap,out,r}})}$
Fan-coil units (V)	$\dot{m}_{\mathrm{air,in}} = \dot{m}_{\mathrm{air,out}} = \dot{m}_{\mathrm{air}}$ $\dot{m}_{\mathrm{fc,in,w}} = \dot{m}_{\mathrm{fc,out,w}} = \dot{m}_{\mathrm{w}}$	$\dot{Q}_{\mathrm{fc}} = \dot{m}_{\mathrm{air}}C_{p,\mathrm{air}}(T_{\mathrm{out,air}} - T_{\mathrm{in,air}})$ $\dot{Q}_{\mathrm{fc}} = \dot{Q}_{\mathrm{cond}};\quad \dot{Q}_{\mathrm{sh}} = \dot{Q}_{\mathrm{cond}}$	$\dot{I}_{\mathrm{fc}} = \dot{m}_{\mathrm{w}}(\psi_{\mathrm{fc,in}} - \psi_{\mathrm{fc,out}}) - \dot{Q}_{\mathrm{fc}}\left(1 - \frac{T_0}{T_{\mathrm{in,air}}}\right)$	$\varepsilon_{\mathrm{fc}} = \frac{\dot{Q}_{\mathrm{fc}}(1-(T_0/T_{\mathrm{in,air}}))}{\dot{m}_{\mathrm{w}}(\psi_{\mathrm{fc,in}} - \psi_{\mathrm{fc,out}})}$
Fan (VI)	$\dot{m}_{\mathrm{fan,in}} = \dot{m}_{\mathrm{fan,out}} = \dot{m}_{\mathrm{fan,air}}$	$\dot{W}_{\mathrm{fan}} = \dot{m}_{\mathrm{fan,air}}(h_{\mathrm{fan,out}} - h_{\mathrm{fan,in}})$	$\dot{I}_{\mathrm{fan}} = \dot{m}_{\mathrm{fan,air}}(\psi_{\mathrm{fan,in}} - \psi_{\mathrm{fan,out}}) + \dot{W}_{\mathrm{fan}}$	$\varepsilon_{\mathrm{fan}} = \frac{\dot{E}x_{\mathrm{fan,out}} - \dot{E}x_{\mathrm{fan,in}}}{\dot{W}_{\mathrm{fan}}}$
Ground heat exchanger (VII)	$\dot{m}_{\mathrm{ghe,in}} = \dot{m}_{\mathrm{ghe,out}} = \dot{m}_{\mathrm{ghe,w}}$	$\dot{Q}_{\mathrm{ghe}} = \dot{m}_{\mathrm{ghe,w}}(h_{\mathrm{ghe,in}} - h_{\mathrm{ghe,out}})$ $\dot{Q}_{\mathrm{sh}} = \dot{Q}_{\mathrm{fc}}$	$\dot{I}_{\mathrm{ghe}} = \dot{m}_{\mathrm{ghe,w}}(\psi_{\mathrm{ghe,out}} - \psi_{\mathrm{ghe,in}}) + \dot{Q}_{\mathrm{ghe}}\left(1 - \frac{T_0}{T_{\mathrm{ghe}}}\right)$	$\varepsilon_{\mathrm{ghe}} = \frac{\dot{E}x_{\mathrm{ghe,in}}}{\dot{E}x_{\mathrm{ghe,out}} + \dot{Q}_{\mathrm{ghe}}(1 - (T_0/T_{\mathrm{ghe}}))} = \frac{\dot{m}_{\mathrm{ghe,w}}\psi_{\mathrm{ghe,in}}}{\dot{m}_{\mathrm{ghe,w}}\psi_{\mathrm{ghe,out}} + \dot{Q}_{\mathrm{ghe}}(1 - (T_0/T_{\mathrm{ghe}}))}$
Solar collector (VIII)	$\dot{m}_{\mathrm{scol,in}} = \dot{m}_{\mathrm{scol,out}} = \dot{m}_{\mathrm{scol,r}}$ $\dot{m}_{\mathrm{scol,in}} = \dot{m}_{\mathrm{scol,out}} = \dot{m}_{\mathrm{scol,w}}$	$\dot{Q}_{\mathrm{w}} = \dot{m}_{\mathrm{scol,w}}C_{p,\mathrm{w}}(T_{\mathrm{out,w}} - T_{\mathrm{in,w}})$ $\dot{Q}_{\mathrm{r}} = \dot{m}_{\mathrm{scol,r}}(h_{\mathrm{scol,out}} - h_{\mathrm{scol,in}})$	$\dot{E}x_{\mathrm{dest,scol}} = \dot{E}x_{\mathrm{u}} + \dot{E}x_{\mathrm{scol}}$ $\dot{E}x_{\mathrm{u}} = \dot{m}_{\mathrm{scol,w}}C_{\mathrm{scol,w}}\left[(T_{\mathrm{out,w}} - T_{\mathrm{in,w}}) - T_0\left(\ln\frac{T_{\mathrm{out,w}}}{T_{\mathrm{in,w}}}\right)\right]$ $\dot{E}x_{\mathrm{scol}} = AI_{\mathrm{T}}\left[1 + \frac{1}{3}\left(\frac{T_0}{T_{\mathrm{sr}}}\right)^4 - \frac{4}{3}\left(\frac{T_0}{T_{\mathrm{sr}}}\right)\right]$	$\varepsilon_{\mathrm{scol}} = \frac{\dot{E}x_{\mathrm{u}}}{\dot{E}x_{\mathrm{scol}}}$
Domestic hot water tank (IX)	$\dot{m}_{\mathrm{dhwt,in}} = \dot{m}_{\mathrm{dhwt,out}} = \dot{m}_{\mathrm{dhwt,w}}$ $\dot{m}_{\mathrm{load,in,w}} = \dot{m}_{\mathrm{load,out,w}} = \dot{m}_{\mathrm{load,w}}$	$\dot{Q}_{\mathrm{dhwt,w}} = \dot{Q}_{\mathrm{dhwt,load}}$ $\dot{Q}_{\mathrm{dhwt,w}} = \dot{m}_{\mathrm{dhwt,w}}(h_{\mathrm{dhwt,in,w}} - h_{\mathrm{dhwt,out,w}})$ $\dot{Q}_{\mathrm{load,w}} = \dot{m}_{\mathrm{load,w}}C_{\mathrm{p,load,w}}(T_{\mathrm{load,in,w}} - T_{\mathrm{load,out,w}})$	$\dot{E}x_{\mathrm{dest,dhwt}} = (\dot{E}x_{\mathrm{dhwt,in}} + \dot{E}x_{\mathrm{load,in}}) - (\dot{E}x_{\mathrm{dhwt,out}} + \dot{E}x_{\mathrm{load,out}})$	$\varepsilon_{\mathrm{dhwt}} = \frac{\dot{E}x_{\mathrm{load,out}} - \dot{E}x_{\mathrm{load,in}}}{\dot{E}x_{\mathrm{dhwt,in}} - \dot{E}x_{\mathrm{dhwt,out}}} = \frac{\dot{m}_{\mathrm{load,w}}(\psi_{\mathrm{load,out}} - \psi_{\mathrm{load,in}})}{\dot{m}_{\mathrm{dhwt,w}}(\psi_{\mathrm{dhwt,in}} - \psi_{\mathrm{dhwt,out}})}$
Floor heating system (X)	$\dot{m}_{\mathrm{fhs,in}} = \dot{m}_{\mathrm{fhs,out}} = \dot{m}_{\mathrm{fhs,w}}$	$\dot{Q}_{\mathrm{fhs}} = \dot{m}_{\mathrm{fhs,w}}(h_{\mathrm{fhs,in}} - h_{\mathrm{fhs,out}})$	$\dot{E}x_{\mathrm{dest,fhs}} = \dot{m}_{\mathrm{fhs,w}}(\psi_{\mathrm{in}} - \psi_{\mathrm{out}}) - \dot{Q}_{\mathrm{fhs}}\left(1 - \frac{T_0}{T_{\mathrm{fhs}}}\right)$	$\varepsilon_{\mathrm{fhs}} = \frac{\dot{Q}_{\mathrm{fhs}}(1 - (T_0/T_{\mathrm{fhs}}))}{\dot{E}x_{\mathrm{in}} - \dot{E}x_{\mathrm{out}}} = \frac{\dot{Q}_{\mathrm{fhs}}(1 - (T_0/T_{\mathrm{fhs}}))}{\dot{m}_{\mathrm{fhs,w}}(\psi_{\mathrm{in}} - \psi_{\mathrm{out}})}$
Circulating pumps (XI)	$\dot{m}_{\mathrm{pump,in}} = \dot{m}_{\mathrm{pump,out}} = \dot{m}_{\mathrm{pump,w}}$	$\dot{W}_{\mathrm{pump}} = \dot{m}_{\mathrm{pump,w}}(h_{\mathrm{out,act}} - h_{\mathrm{in}})$ $\dot{W}_{\mathrm{pump,elec}} = \dot{W}_{\mathrm{pump}}/(\eta_{\mathrm{pump,elec}}\eta_{\mathrm{pump,mech}})$ $\dot{W}_{\mathrm{pump,elec}} = V_{\mathrm{pump}}I_{\mathrm{pump}}\cos\varphi$	$\dot{E}x_{\mathrm{dest,pump}} = \dot{m}_{\mathrm{pump,w}}(\psi_{\mathrm{in}} - \psi_{\mathrm{out}}) + \dot{W}_{\mathrm{pump,elec}}$	$\varepsilon_{\mathrm{pump}} = \frac{\dot{E}x_{\mathrm{out}} - \dot{E}x_{\mathrm{in}}}{\dot{W}_{\mathrm{pump,elec}}} = \frac{\dot{m}_{\mathrm{pump,w}}(\psi_{\mathrm{out}} - \psi_{\mathrm{in}})}{\dot{W}_{\mathrm{pump,elec}}}$

• Productivity lack:

$$\xi_i = \frac{\dot{I}_i}{\dot{P}_T} \quad (11)$$

• Exergetic factor:

$$f_i = \frac{\dot{F}_i}{\dot{F}_T} \quad (12)$$

6. Conclusions

HPWHs, unlike conventional WHs that use either gas burners (and sometimes other fuels) or electric resistance heating coils to heat the water, are devices, which operate on an electrically driven vapor-compression cycle and pump energy from the air in its surroundings to water in a storage tank, thus raising the temperature of the water. Its primary advantage is a significant increase in point-of-use operating efficiency over alternative electric water-heating systems [44].

With water heating the second largest use of home energy in most locations, there has been renewed interest in the development and use of energy-conserving domestic WHs since the 1973–1974 oil embargo [45].

The main conclusions, which may be drawn from the results of the present study, are listed below:

(a) This study reviewed HPWHs in a period from 1976 to 2007. HPWHs were classified in four main groups: GSHP; ASHPs; solar-assisted HPs and GEHP.
(b) Both theoretical and experimental analyses done were much in numbers.
(c) Energy analysis method was used in many studies, while the number of studies conducted on exergy analysis was very low.
(d) Since space heating and A/C applications have more requirements, many of the systems are to be used in water heating applications.
(e) The performance indicator of GSHPs, namely COP, ranged from 1.656 to 3.307, considering cooling or heating aims. For ASHPs, these values changed from 1.8 to 5.66, according to ambient temperature, internal–external and water heating.
(f) Solar-assisted HP studies used different collectors, its efficiency and COPs varied from 0.08 to 1.08 and from 1.7 to 6, respectively. Only one study included exergy analysis, while its exergetic system efficiency and COP were 44.06% and 0.201, respectively.
(g) GEHP systems have become more efficient when used both in water and space heating.
(h) When HPWHs are compared with other WHs, HPWHs are second after solar with electric back-up. Their energy savings minimum standards, expected energy savings over equipment life time and expected life time are 65%, up to 900 (US$) and 10 years.
(i) HPWHs have considerable promise for use in both residential and commercial applications. Residential HPWH units have been available for more than 20 years, but have experienced limited success in the marketplace. [1].
(j) Commercial-scale HPWHs are a very promising technology. However, their present market share is extremely low, and only two or three manufacturers are seriously involved in the market. The market needs further conditioning for successful market transformation. The next step in this market conditioning is likely to be further demonstrations and developing additional, well-documented case studies that will support sales efforts to designers and customers [1].
(k) At current production volumes, HPWH prices are beyond customers' likely willingness to pay, preventing widespread implementation of the product. This research indicates that the future of HPWHs is likely to be consistent with the past unless they are produced and sold in large enough quantities to reduce prices and increase availability. Policies or programs to promote efficient WHs may be necessary to make that happen [2].
(l) By 2020, annual electricity savings from HPWHs could be equal to between 1% and 2% (i.e., 12,000 to 24,000 GW h) of total current residential electricity consumption in the U.S. That assumes an average national electric WH market share of a little less than 20% to a little less than 40% (or roughly 25% to 50% of its ‘‘target market’’) over the 2007 through 2020 time period [2].

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References

[1] Kokkinides L, Sachs HM. Toward market transformation: commercial heat pump water heaters for the ‘‘New York energy $marts Region’’. Final Report. Prepared for The New York State energy research and development authority and prepared by American council for an energy—efficient economy, NYSERDA, October 2002, http://aceee.org/buildings/coml_equp/hpw/hpwhnyes.pdf and www.aceee.org/buildings/coml_equp/hpwh/cdhrpt.pdf [Access date: January 15, 2008].
[2] Harris J, Neme C, Calwell C. Residential heat pump water heaters: energy efficiency potential and industry status. Prepared for Natural Resources Defense Council, November 2005. http://www.energystarpartners.net/ia/Water_heaters/Documents/VEIC_HPWHfinalreport_Nov2005.pdf [Access date: January 27, 2008].
[3] Kim M, Kim MS, Chung JD. Transient thermal behavior of a water heater system driven by a heat pump. International Journal of Refrigeration 2004;27:415–21.
[4] Kara O, Ulgen K, Hepbasli A. Exergetic assessment of direct-expansion solar-assisted heat pump systems: review and modeling. Renewable and Sustainable Energy Reviews 2008;12(5):1383–401.
[5] Zhang J, Wang RZ, Wu JY. System optimization and experimental research on air source heat pump water heater. Applied Thermal Engineering 2007;27:1029–35.
[6] Luo Q, Tang G, Liu Z, Wang J. A novel water heater integrating thermoelectric heat pump with separating thermosiphon. Applied Thermal Engineering 2005;25:2193–203.
[7] Neksa P, Rekstad H, Zakeri GR, Schiefloe PA. CO_2-heat pump water heater: characteristics, system design and experimental results. International Journal of Refrigeration 1998;21:172–9.
[8] Laipradit P, Tainsuwan J, Kiatsiriroat T, Aye L. Theoretical performance analysis of heat pump water heaters using carbon dioxide as refrigerant. International Journal of Energy Research 2008;32:356–66.
[9] Rankin R, Rousseau PG, Eldik MV. Demand side management for commercial buildings using an inline heat pump water heating methodology. Energy Conversion and Management 2004;45:1553–63.
[10] Rousseau PG, Greyvenstein GP. Enhancing the impact of heat pump water heaters in the South African commercial sector. Energy 2000;25:51–70.
[11] U.S. Department of Energy. Energy Source. Geothermal. History of Geothermal Energy in the United States. Geothermal Technologies Program. http://www1.eere.energy.gov/geothermal/history.html [Accessed date: September 18, 2007].
[12] The Equitable Building Heat Pump System. http://files.asme.org/ASMEORG/Communities/History/Landmarks/5541.pdf [Accessed date: January 20, 2007].
[13] The Equitable Building Heat Pump System. A National Historic Mechanical Engineering Landmar. Dedicated May 8, 1980 in Portland, Oregon. http://www.bookrags.com/research/heat-pump-woi/ [Access date: January 25, 2008]
[14] The unitary heat pump industry—25 years of progress. http://adsabs.harvard.edu/abs/1977ASHRA.19.15P [Access date: September 19 2007].
[15] Analysis of Geothermal Heat Pump Manufacturers survey data. http://www.eia.doe.gov/cneaf/solar.renewables/rea_issues/geo_hp_art.pdf [Accessed on September 15, 2007].
[16] Swardt CAD, Meyer JP. A performance comparison between an air-source and a ground-source reversible heat pump. International Journal of Energy Research 2001;25:899–910.

[17] Hepbasli A, Akdemir O, Hancioglu E. Experimental study of a closed loop vertical ground source heat pump system. Energy Conversion and Management 2003;44:527–48.
[18] Biaou AL, Bernier MA. Achieving total domestic hot water production with renewable energy. Building and Environment 2008;43:651–60.
[19] Sloane BD, Krise RC, Kent DD. Energy Utilization Systems, Inc., Demonstration of a heat pump water heater, A subcontracted report, ORNL/Sub-7321/3, December 1979.
[20] Harata I, Takagi S, Wakabayashi I, Sakai M, Tetuka H, Toshou T. The development of the storage thank water heater by thermoelectric technology. In: Proceedings of the 17th international conference on thermoelectrics, ICT; 1998. p. 547–50.
[21] Ito S, Miura N. Studies of a heat pump using water and air heat sources in parallel. Heat Transfer Asian Research 2000;29:473–90.
[22] Ji J, Chow T, Pei G, Dong J, He W. Domestic air-conditioner and integrated water heater for subtropical climate. Applied Thermal Engineering 2003; 23:581–92.
[23] Morrison G, Anderson T, Behnia M. Seasonal performance rating of heat pump water heaters. Solar Energy 2004;76:147–52.
[24] Sakai I, Terakawa MT, Ohue J. Solar space heating and cooling with bi-heat source heat pump and hot water supply system. Solar Energy 1976;18: 525–32.
[25] Chaturvedi SK, Chen DT, Kheireddne A., Thermal performance of a variable capacity direct expansion solar-assisted heat pump. PII: S0196-8904(96)00228-2.
[26] Chaichana C, Aye L, Charters WWS. Natural working fluids for solar-boosted heat pumps. International Journal of Refrigeration 2003;26:637–43.
[27] Hawlader MNA, Chou SK, Jahangeer KA, Rahman SMA, Eugene Lau KW. Solar-assisted heat-pump dryer and water heater. Applied Energy 2003;74: 185–93.
[28] Chyng JP, Lee CP, Huang BJ. Performance analysis of a solar-assisted heat pump water heater. Solar Energy 2003;33–44.
[29] Kuang YH, Sumathy K, Wang RZ. Study on a direct-expansion solar-assisted heat pump water heating system. International Journal of Energy Research 2003;27:531–48.
[30] Huang BJ, Lee JP, Chyng JP. Heat-pipe enhanced solar-assisted heat pump water heater. Solar Energy 2005;78:375–81.
[31] Guoying X, Xiaosong Z, Shiming D. A simulation study on the operating performance of a solar-air source heat pump water heater. Applied Thermal Engineering 2006;26:1257–65.
[32] Li YW, Wang RZ, Wu JY, Xu YX. Experimental performance analysis on a direct-expansion solar-assisted heat pump water heater. Applied Thermal Engineering 2007;27:2858–68.
[33] Hepbasli A. Exergetic modeling and assessment of solar assisted domestic hot water tank integrated ground-source heat pump systems for residences. Energy and Buildings 2007;39:1211–7.
[34] Trilliant-Berdal V, Souyri B, Fraise G. Experimental study of ground-coupled heat pump combined with thermal solar collectors. Energy and Buildings 2006;38:1477–84.
[35] Lian Z, Park S, Huang W, Baik Y, Yao Y. Conception of combination of gas-engine-driven heat pump and water-loop heat pump system. International Journal of Refrigeration 2005;28:810–9.
[36] Zhang RR, Lu XS, Li SZ, Lin WS, Gu AZ. Analysis on the heating performance of a gas engine driven air to water heat pump based on a steady-state model. Energy Conversion and Management 2005;46:1714–30.
[37] Lazzarin R, Noro M. District heating and gas engine heat pump: economic analysis based on a case study. Applied Thermal Engineering 2006;26:193–9.
[38] Hepbasli A, Erbay Z, Icier F, Colak N, Hancioglu E. A review of gas engine driven heat pumps (GEHPs) for residential and industrial applications. Renewable and Sustainable Energy Reviews, 2007; in press.
[39] High Efficiency Water Heaters Provide Hot Water for Less. http://www.energystar.gov/ia/new_homes/features/WaterHtrs_062906.pdf [Accessed date: October 14, 2007].
[40] Gunerhan H, Hepbasli A. Exergetic modeling and performance evaluation of solar water heating systems for building applications. Energy and Buildings 2007;39:509–16.
[41] Hepbasli A, Akdemir O. Energy and exergy analysis of a ground-source (geothermal) heat pump system. Energy Conversion and Management 2004;45:737–53.
[42] Van Gool. Energy policy: fairly tales and factualities. In: Soares ODD, Martins da Cruz A, Costa Pereira G, Soares IMR, Reis AJP, editors. Innovation and technology—strategies and policies. Dordrecht: Kluwer; 1997. p. 93–105.
[43] Xiang JY, Cali M, Santarelli M. Calculation for physical and chemical exergy of flows in systems elaborating mixed-phase flows and a case study in an IRSOFC Plant. International Journal of Energy Research 2004;28:101–15.
[44] Bernardi WP, Blevins RP, Sloane BD. Examination and life assessment of field-tested heat pump water heaters. Prepared by Energy Utilization Systems, Incorporated for Oak Ridge National Laboratory operated by Union Carbide Corporation for the U.S. Department of Energy, December 1982.
[45] Sauer HJ, Howell RH. Heat pump systems. USA: John Wiley & Sons, Inc.; 1983